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A.S.M.E. Spring Meeting Papers

MECHANICAL ENGINEERING

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A WHITE SULPHUR VISTA

MID-MAY 1927

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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The Purpose of This Issue

THIS special Mid-May issue of MECHANICAL ENGINEERING brings advance information about the Program of the Spring Meeting of the A.S.M.E. at White Sulphur Springs, W. Va., together with the text of a number of complete papers and synopses of others that are available in pamphlet form. Copies of the latter may be obtained by filling in the form on page 591. This supplement, however, should be brought to the Meeting, as copies of the papers appearing in it will not be available for free distribution. However, additional copies of the supplement may be purchased at the Meeting.

It is the purpose of this issue to make available to the members of the A.S.M.E. information regarding the papers that are to be presented, so that discussion at the Meeting will be well considered and thoughtfully prepared. For an author to read a technical paper from beginning to end at a meeting is a waste of valuable time, for the audience is seldom able to evaluate statements that are made and compare them with their own experience.

It has been the effort of the Committee on Meetings and Program to stimulate discussion at the meetings of the Society, knowing that a spirited discussion insures an interesting meeting. A paper which has been subjected to public criticism or augmented by corroboratory information is, when properly recorded, of much greater value as a part of the permanent literature of mechanical engineering than a spiritless or an uncriticized contribution. In some cases the discussion may be of even greater value than the paper. From its value in instilling interest in the meeting and from its effect on the record of the meeting, discussion is therefore vitally essential. The Committee on Meetings and Program and the Committee on Publications have coöperated in this procedure for presenting and issuing papers for the Spring Meeting at White Sulphur Springs.

The Spring Meeting at White Sulphur

WHITE Sulphur Springs, W. Va., far removed from the sights and sounds of any industrial activity, a famous pleasure resort and one of the most beautiful spots in America, has been chosen for the 1927 Spring Meeting of The American Society of Mechanical Engineers. The activities of this resort all center around the Greenbrier and Cottages located upon a vast estate in the Greenbrier Mountains, 2000 feet above sea level. This hotel with its unusual facilities for the technical sessions and for social and sport events will be at the full disposal of the A.S.M.E. members and their families during the Meeting.

The Program for the Meeting is on the opposite page. The current issues of the *A.S.M.E. NEWS* are giving particulars of the many interesting events to be held during the four days of the meeting.

MECHANICAL ENGINEERING

Volume 49 Mid-May, 1927 Number 5a

Spring Meeting Program

Greenbrier Hotel, White Sulphur Springs, W. Va., May 23 to 26, 1927

Monday, May 23

Morning:
Council Meeting

Afternoon:
Council Meeting
Business Meeting, 1:30 P.M.

Afternoon: Simultaneous Sessions: 2:30 P.M.

Education and Training for the Industries

The General Motors Institute of Technology, ALBERT SOBEY.
A Description of the Henry Ford Trade School, F. E. SEARLE.
A Uniform Apprenticeship Certificate for Coöperating Groups of Employers, WILLIAM S. CONANT.

Central Station Power

High-Pressure Steam at Edgar Station, Boston, I. E. MOULTROP and E. W. NORRIS.
High Steam Pressure and Temperature at Crawford Avenue Station, A. D. BAILEY.
Strength of Pipe Flanges, E. O. WATERS and J. HALL TAYLOR.

Tuesday, May 24

Morning: Simultaneous Sessions: 9:30 A.M.

Fuels

Economics of Coal Carbonization in the United States, GEO. A. ORROK.

Management

Accident and Production, L. W. WALLACE.

Wood Industries

(Jointly with Materials Handling Division)
Material Handling Between the Stump and the Board, L. C. BELL.
Design of Motor-Bus Bodies, L. C. JOSEPHS, JR.

Conference Local Sections Delegates: 10 A.M.

Wednesday, May 25

Morning: Simultaneous Sessions: 9:30 A.M.

Hydraulic

Rock Structure and Headgates of Cedar Creek Hydroelectric Station, W. S. LEE.
Specific Characteristics for Hydraulic Turbines, ARNOLD PFAU.
Comparison and Limitations of Various Water-Hammer Theories, R. S. QUICK.
Progress Report of Hydraulic Division.

Machine Shop Practice

Hysteresis Relative to the Operation of Mechanical Springs, J. K. WOOD.
Arc Welding, J. F. LINCOLN.
Vibration of Frames of Electrical Machines, J. P. DEN HARTOG (by title).

Oil and Gas Power

Study of Oil Sprays for Fuel-Injection Engines by Means of High-Speed Motion Pictures, E. G. BEARDSLEY.
Experimental Combustion Chambers Designed for High-Speed Diesel Engines, C. KEMPER.
Some Uses of the High-Speed Multi-Cylinder Indicator, H. M. JACKLIN.
Range of Severity of Torsional Vibration in Diesel Engines, F. P. PORTER (by title).

Thursday, May 26

Morning: Simultaneous Sessions: 9:30 A.M.

Railroad

High Steam Pressures in Locomotive Cylinders, L. H. FRY.
Diesel Traction for Railroads, WM. ARTHUR.

Industrial Power

Management of Industrial Power, Engineer's Viewpoint, H. F. SCOTT.
Management of Industrial Power, Manager's Viewpoint. (Author to be announced.)
Evaporators for Boiler Feed Make-Up Water, W. L. BADGER.

Materials Handling

The Relation of Building Design to the Manufacturing Process, CHAS. P. WOOD.
Actual Layout of a Building Around a Materials-Handling Installation. (Author to be announced.)

Entertainment Events

Monday, May 23, *Evening:*
Moving pictures.
Dancing.

Tuesday, May 24, *Afternoon:*
Golf and Tennis Tournaments. Motor Drive and Tea for the Ladies.

Evening:
Bridge Tournament simultaneous with movies, followed by dancing.

Wednesday, May 25, *Afternoon:*
Ball game between A.S.M.E. and A.S.R.E.
Golf and Tennis Tournaments.
Putting Contest for Ladies.

Evening:
Dinner and award of A.S.M.E. Medal to WILFRED LEWIS. Address by LOUGHNAN S. PENDRED, Editor, *The Engineer*, London, England.
Award of tournament prizes.
Bridge Party and Water Carnival.

Thursday, May 26, *Afternoon:*
Golf.

General View of Modern Sawmill Plant in the Heart of the Appalachians, Showing Sawmill, Planing Mills, Dry Kilns, Lumber Yard, Boarding House, Club House, Residences, Etc.

Material Handling Between Stump and Board

Processes Employed in the Appalachian Mountain Hardwood Section—Logging Railroads—Felling Trees—"Ballhooting"—Skidding—Loading Log Trains—Log Ponds—Handling in the Mill—Piling and Stacking—Loading Lumber on Cars, Etc.

By LANDON C. BELL,[1] COLUMBUS, OHIO

THE scope of this paper is indicated by the title, but the title needs some limitations, for it is proposed to confine the discussion to material handling in the Appalachian hardwood section. This, in general, is a mountainous territory, where the problems are different from those encountered in lowland and swamp logging.

Preliminary

Preliminary to a description of the actual processes employed, it will serve a purpose to notice briefly the physical characteristics of the country to which the discussion relates.

The hardwood-lumber industry in the states of West Virginia, Virginia, Kentucky, eastern Tennessee, and western North Carolina—the territory usually referred to as the Appalachian hardwood area—is, so far as the producing branch of that industry is concerned, confined largely to the mountainous areas. This is due to the fact that the flat lands have in the years gone by been largely cleared for agriculture. The forest growths have remained in the mountains because the lands were not adapted to agriculture, and because they were inaccessible. It was impossible in the past to utilize the fine hardwood timber growing in the remote recesses of the Appalachian Mountains, because the cost of getting to the trees, cutting the timber, and getting the logs to sawmills and lumber to market was so great that it was prohibitive.

It is impossible to give one who has never been in the wilds of these mountains an adequate idea of the magnitude of the scale upon which nature has constructed in these mountains her grand panorama of diversified physical features of barbarity and splendor. The mountain ranges, the secondary ridges, thrown up to varying elevations running in every direction, intersecting each other, contending for the mastery of control not only of elevation but of the flow of the streams; rugged, almost perpendicular cliffs, abrupt and sloping mountain sides, narrow valleys with scarcely an acre of level land, almost impenetrable ravines; streams in dry weather mere rivulets, at other times mountain torrents with the destructive power of a cataract, tumbling down the mountain sides with such steep descent as to approach at times the character of waterfalls, are among the principal elements which shape the problems of material handling in these sections. Fig. 1 gives some idea of the rough and rugged character of much of the section discussed.

In this territory the usual method of operation is to select the location of the plant site, and make it the focus of all operations. Usually the difficulty is to find sufficient land relatively level upon which to locate the necessary facilities, such as saw mill, log pond, lumber yard, etc.

The plant site must be selected, of course, not only with a view to providing sufficient space for such facilities as have been mentioned, but it must also be so located that it is accessible to the timber supply which is to provide its raw material, and at the same time it must have shipping facilities for the lumber produced from the trees.

With these brief observations intended to give some idea of the facts and circumstances which determine the essential character

[1] W. M. Ritter Lumber Co.

Contributed by the Wood Industries Division for presentation at the Spring Meeting, White Sulphur Springs, W. Va., May 23 to 26, 1927, of The American Society of Mechanical Engineers, 29 West 39th St., New York. All papers are subject to revision.

Be sure to bring this copy with you to the Spring Meeting

of the problems in handling hardwood-forest products from the tree to the board, we may attempt a description of the processes involved.

The Trees

We may begin with the trees themselves. Fig. 2 gives a very good idea of the trees from which the material to be handled is produced. The illustration is made from a photograph taken where the ground is more nearly level and the stand of timber somewhat denser than the average. In the Appalachian highlands the average stand of merchantable timber in virgin forests is about 5000 feet per acre. Of course it is not uniform. Sometimes the

FIG. 1 VIEW SHOWING ROUGH AND RUGGED CHARACTER OF THE APPALACHIAN HARDWOOD SECTION

FIG. 2 REPRESENTATIVE STAND OF TIMBER (WHITE OAK)

stand is less than half that figure, sometimes more than twice that much.

Logging Railroads

It may be interesting to know the magnitude of the material-handling problem in a modern Appalachian hardwood-lumber operation. This may be indicated by reference to some of the figures for a representative hardwood operation in that area.

FIG. 3 ROCK CUT FOR LOGGING ROAD

FIG. 4 CHARACTER OF TRESTLE WORK OFTEN REQUIRED

The material—the logs, about 450,000,000 ft.—was produced from an area aggregating 110,000 acres; in order to consolidate that acreage into an operative proposition it was necessary to purchase 1073 separate tracts from over 1200 different persons.

To operate the property requires 200 miles of railroad, but only about 40 miles are needed at any one time. The cost of building the railroad is about $2.45 per 1000 ft. B. M. of the material, exclusive of the rails.

Thus it is seen that the problems of getting the material to move and of providing the primary means for moving it, while outside

FIG. 5 FELLING A TREE

FIG. 6 LOGGERS FREEING A PARTLY PEELED LOG PRIOR TO "BALLHOOTING" IT DOWN THE MOUNTAIN SIDE

FIG. 7 SKIDDING LOGS WITH TEAMS

the immediate scope of this paper, are problems of the first magnitude for the lumberman.

We get an inadequate idea of the railroad problems from the accompanying illustrations, but they serve to illustrate the difficulties of one phase of this material-handling work.

Fig. 3 shows a pretty expensive rock cut for a logging road, which at most can have but a relatively short life.

Fig. 4 shows the character of trestle work often necessary to be employed.

FELLING THE TREES—"BALLHOOTING"

When the forests have been purchased, the mill plant built, and the railroad constructed from sawmill to woods, the enterprise is ready for the material handling between the tree and board to begin. The first step is the felling of the trees. Two woodsmen tackling this job are shown in Fig. 5.

In Fig. 6 is shown part of a large tree that has been sawed into logs, on a steep mountain side. The territory where these trees grow is often so steep and inaccessible that neither railroads, tractors, nor even horse teams can reach them. In such cases the logs are partly peeled, in order that the peeled surface may offer little frictional resistance; the ends of the logs are scalped or rounded, and they are then turned loose, end foremost, and allowed to slide by their own momentum down the mountain side. This process is known in the language of the lumberjack as "ballhooting."

The figure shows the loggers, with the aid of peavies or cant

FIG. 8 SNAKING TRAIL OVER RAVINE

FIG. 9 SNAKING LOGS WITH A TRACTOR

hooks, freeing the lower end of one log from contact with the next one so it can be started down the mountain side.

Needless to say, the cost of that part of the handling process from the time the log is released at the top of a steep mountain until it comes to rest far below, on ground so relatively level that

it cannot travel farther by its own momentum, is not very great.

Fig. 7 shows logs being picked up after their wild career of "ball-hooting" down the mountain side, by skidding crews with horses for another lap of their journey to the sawmill.

The necessity of employing several different methods of transportation in getting the logs from the forests to the mills in the

FIG. 10 STEAM SKIDDER AT WORK

FIG. 11 LOADING LOGS ON LOGGING CARS

Appalachian hardwood area arises mainly from topographical considerations. The ideal plan would be to load the logs directly upon the log train and haul them to the mill. But it is not practical to operate railroads where the grade would be too steep; switchbacks are often necessary, but they are expensive, and cannot be resorted to except on the semi-permanent stretches of the road. Likewise tractors, because of the steepness and roughness of many sections, cannot be employed; they are not practical even in areas where logging teams of horses can be used advantageously. In some sections mechanical steam-driven skidders can be employed to advantage, and in some circumstances steam skidding is about the only practical method that can be employed.

But whether the means employed be teams of horses, tractors, steam skidders, or a combination of these, the aim and object of the logger is to get the logs from the forest assembled at points accessible to the logging railroad so that they may be loaded upon the log train.

The difficulty and expense of making slides or snaking trails over which horses may draw single files of logs is shown in Fig. 8. Here the trail had to be constructed over a ravine-like depression which was filled up with logs.

Where the topography is such as to render the use of tractors available, it is practicable to employ the method of "snaking" in logs shown in Fig. 9. This method, however, is not in general use in the Appalachian highlands.

The method of bringing in the logs by skidding machinery is illustrated in Fig. 10, which shows a steam skidder at work.

Whatever the method of skidding employed, the object is to assemble the logs in log yards or "jackpots," so they can be loaded upon the log train. The term "jackpot" used in this connection has its own peculiar meaning, and is used to designate a large concentration of logs accessible to the logging railroad. Sometimes thousands of logs are dumped into depressions or hollows adjacent to the logging railroad, and present much the appearance of an enormous pile of giant jackstraws (see Fig. 10).

LOADING

When the logs have reached the "jackpot" or have been assembled by the logging road, they are ready to be loaded on the logging cars. This step in the handling of the material is accomplished almost exclusively by means of the steam loaders. Figs. 11 and 12 show the loading process.

Fig. 12 is interesting in that it shows several details illustrating

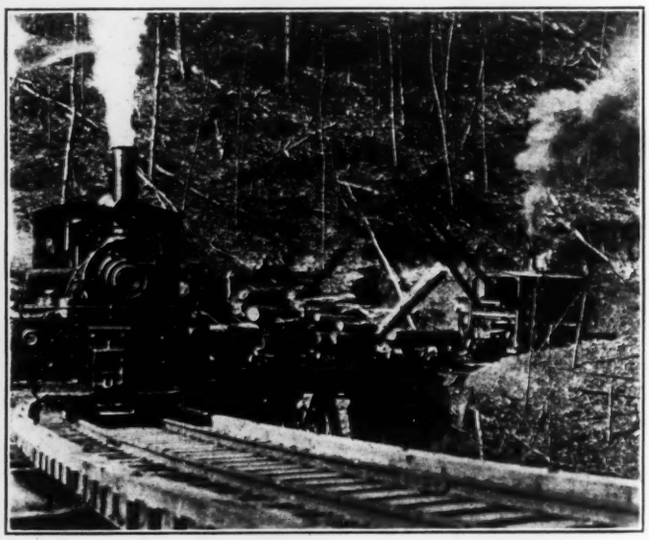

FIG. 12 LOADING LOGS ON LOGGING CARS

FIG. 13 SWITCHBACK ON LOGGING RAILROAD

the difficulties of Appalachian hardwood logging. The logs being loaded were assembled from the territory on the side of the stream where the loader is stationed. These were brought to the assembling point from the territory to the left of the picture. In order to reach these logs, the bridge or trestle shown in the fore-

ground had to be built, and in order to get space sufficient for the loader to be set, no inconsiderable excavation had to be made into the mountain side.

THE LOG TRAINS

The loaded log cars are made up into train loads. The number of cars that can be pulled from the woods to the mill varies so greatly on account of the topography of different sections that no general rule can be stated. The load varies somewhat, dependent upon whether broad-gage or narrow-gage equipment is used. Broadly speaking, loads may be said to vary from ten to thirty loaded cars.

In discussing logging-railroad construction, some mention has been made of the difficulties and expense involved. The illustrations given above in that connection convey an inadequate idea of the matter, and additional insight into it is afforded by some of those to follow, showing the log trains.

Fig. 13 shows a piece of switchback construction, occurring where trestle work on both prongs is required.

Fig. 14 shows a long train of fine Appalachian hardwood logs on a stretch of logging road built across a river and over a railroad (the Carolina, Clinchfield & Ohio).

THE LOG POND AND JACK SLIP

The logs, when brought in from the woods to the mill, are dumped into the log pond. The log pond is a feature of all large hardwood plants in the Appalachians. It serves several purposes, being the concentration depot for the logs at the mill, where they can be assembled, sorted, cleaned, and fed directly into the mill.

Fig. 15 shows a log pond full of logs. It also shows the end of the mill building, with the jack slip, the incline arrangement leading from the log pond up into the mill. Pond men on a float guide the logs into the jack slip, up which they are carried to the log deck by means of an endless chain with barbed teeth.

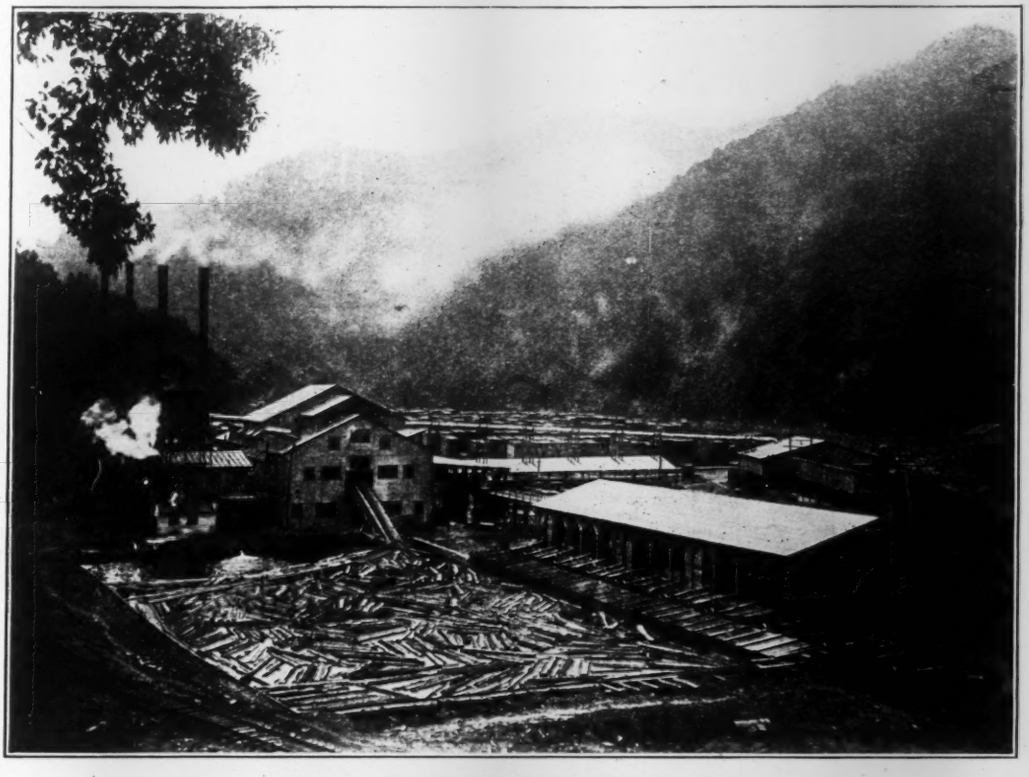

FIG. 15 MILL WITH LOG POND IN FOREGROUND

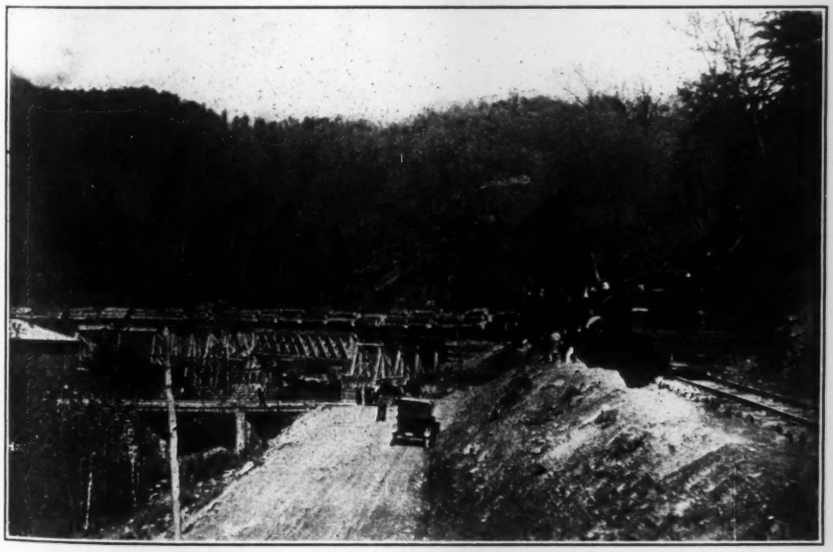

FIG. 14 LONG TRAIN OF HARDWOOD LOGS ON ROAD BUILT OVER A RIVER AND A RAILROAD

HANDLING IN THE MILL

From the jack slip the logs are delivered to the log deck, and thence are received on the movable carriage upon which the log is held fast while the sawing process takes place.

The log deck is located in the far end of the mill interior as shown in Fig. 16. In the foreground of this illustration may be seen the edging machines and cut-off tables, as well as the sorting tables with their endless-conveyor chains.

On the left of this picture may be seen the live rolls, which convey the lumber not necessary to be edged or cut off to the proper pair of conveyor chains.

Fig. 17 shows a close-up of an operative receiving a sawed-and-edged board upon the table which takes it to the cut-off saw. Groups of saws operated automatically upon disappearing drums are at the command of this man, who can, by simply stepping on a lever, cut off the board to any desired length.

Fig. 18 shows a grader and tally man inspecting the boards as they pass from the sawmill, after being completely manufactured, sawed, edged, trimmed, and cut off, to the trucks to be taken to the yard and stacked for drying.

The work of these men is to designate the place where every board will go in the yard. In a typical hardwood plant about 150 different kinds, sizes, thicknesses, and grades of lumber are produced, thus making the sorting and assembling of the miscellaneous production of the mill in the yard a most important and exacting matter.

The lumber thus sorted is loaded on small lumber trucks, which usually are mounted on flanged wheels and run on a miniature steel rail. These are often simply pushed from the mill to the lumber yard by hand, the platforms upon which the trucks are laid being carefully graded for easy conveyance of the load. However, it often happens that on account of the distance or weight of the load, it is found desirable to use the faithful mule for pulling the lumber buggies.

Piling and Stacking

When the lumber is conveyed from the mill to the yard it is stacked in separate piles according to the species, thickness, grade, etc., to dry. Ordinarily this process requires from four to seven months, depending upon the kind of lumber and the thickness of the board.

Fig. 19 gives a good idea of a yard arrangement and piling plan, while Fig. 20 shows the method of building up the individual pile.

Fig. 21 shows the care and precision employed in piling the lumber and placing the stickers so that drying conditions will be such that the finished product will be as uniform as possible.

Fig. 22 shows the method of piling and preserving small pieces of boards and lumber known as "dimension stock." This requires special care, and it is piled and preserved under cover as can be seen in the cut.

After the lumber has been piled a sufficient length of time or kiln dried, as the case may be, the final stage, so far as the sawmill man is concerned, is the shipping process.

Loading on Cars

Fig. 23 shows men at work unstacking and inspecting lumber to be loaded in railroad cars, as shown in Fig. 24. In Fig. 23 it will be seen that the end of the pile has been boarded up with rough lumber. This was done to reduce splitting and checking as much as possible.

Although the lumber is graded green as it comes from the saw and is thus piled in the yard, it has to be inspected again when shipped because a certain amount develops defects and imper-

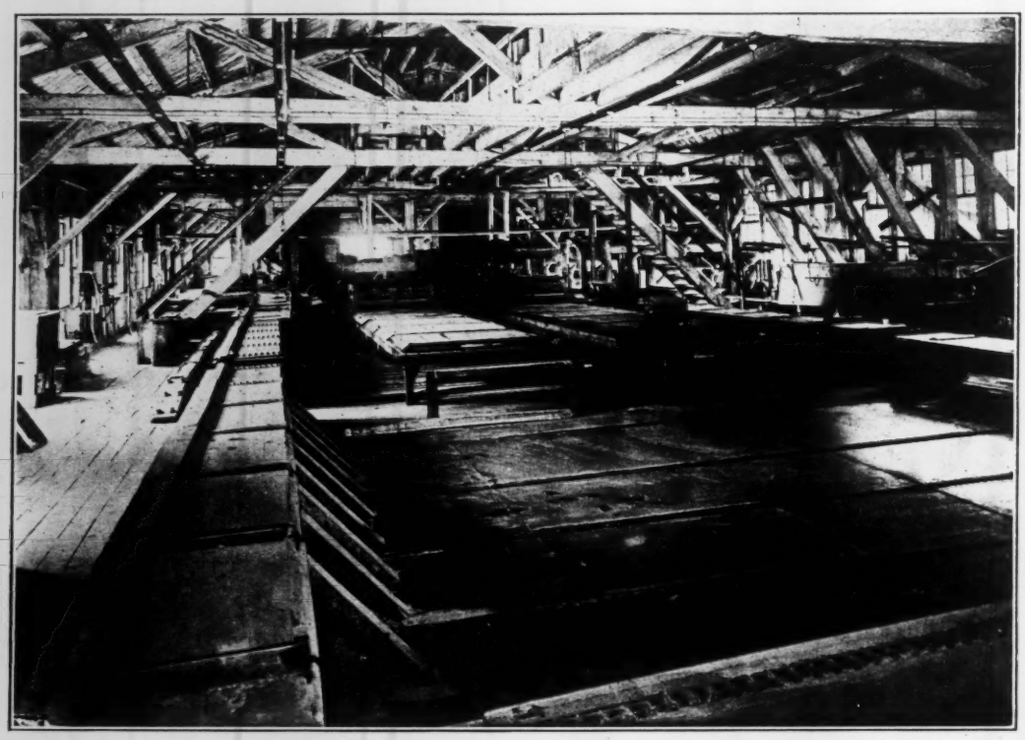

Fig. 16 Interior of Mill

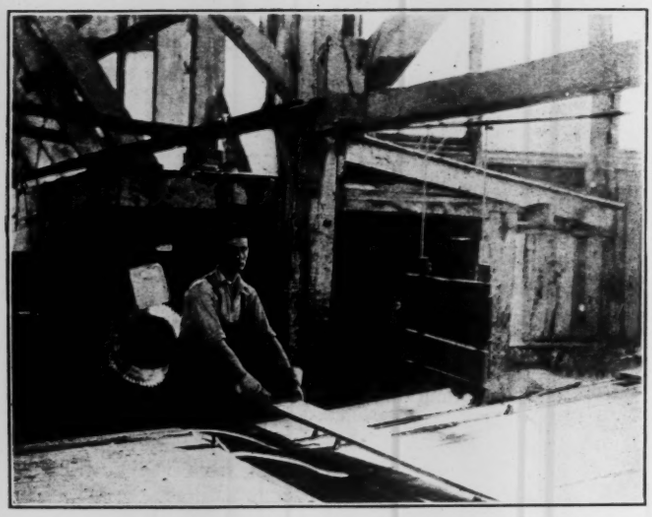

Fig. 17 Operative Receiving Sawed-and-Edged Board at Table Which Takes It to Cut-Off Saw

Fig. 18 Grader and Tally Man Inspecting Manufactured Lumber

fections in drying and seasoning which were not apparent upon the inspection in the green state.

The defective ends are cut off by two men with a cross-cut saw before the boards are loaded into the car. This is done to eliminate defects found upon the final inspection of the dry lumber preparatory to being loaded for shipment. Incidentally this illustrates the wastage of a lot of lumber good for all practical purposes, but which the more or less unscientific lumber-grading rules make imperative. It is such wastes as this that the Simplification and Standardization of Grading Rules will do much to prevent.

It is thus seen that there are many stages (and all are not reviewed above) in the process of handling the products of the

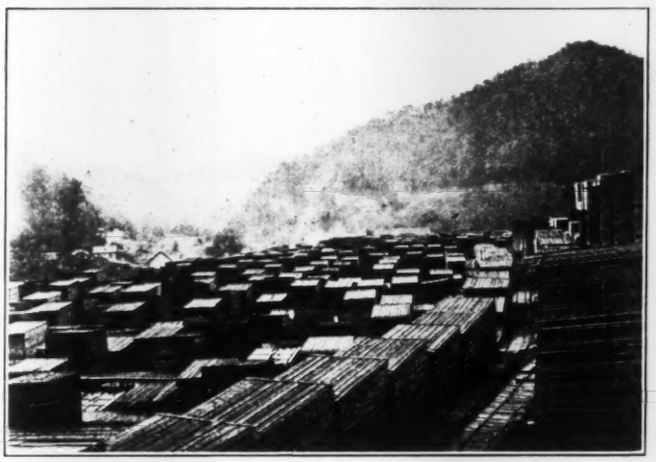

FIG. 19 YARD ARRANGEMENT AND PILING PLAN

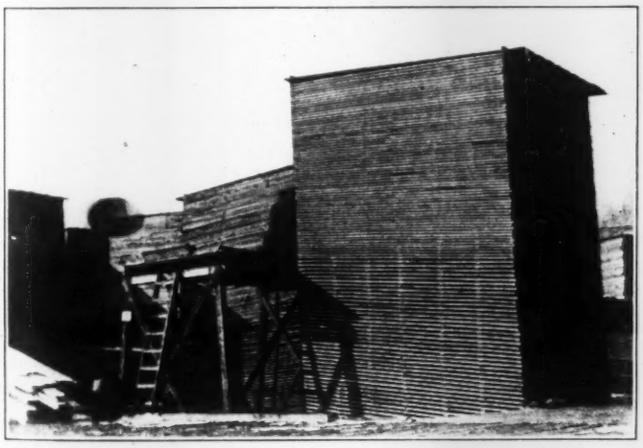

FIG. 21 SHOWING CARE AND PRECISION EMPLOYED IN PILING LUMBER SO THAT DRYING WILL BE UNIFORM

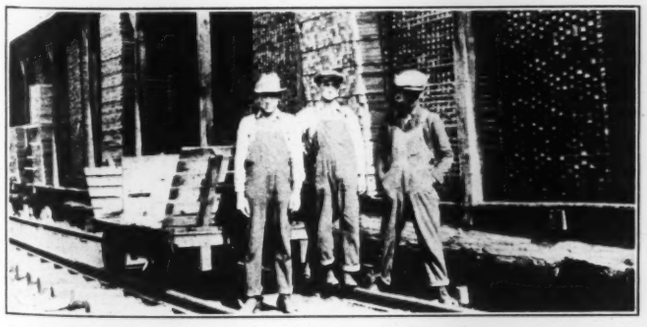

FIG. 22 METHOD OF PILING AND PRESERVING DIMENSION STOCK

Appalachian hardwood forests from the stump until the lumber is in the railroad car, on its way to the consumer.

The living, eating, and sleeping quarters of the woodsmen—"lumberjacks" as they are familiarly and affectionately called—are in specially constructed camp cars, built on railway trucks, and capable of being moved from place to place and installed for any desirable length of time upon sidings of the lumber railroads.

The illustration at the beginning of the paper presents as much as can be shown in one photograph of a modern sawmill plant in

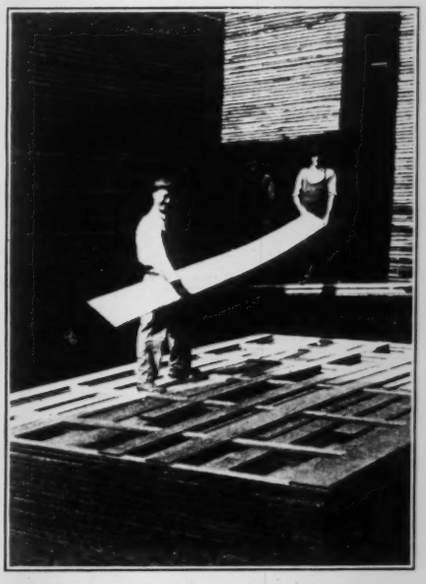

FIG. 20 METHOD OF BUILDING UP AN INDIVIDUAL PILE

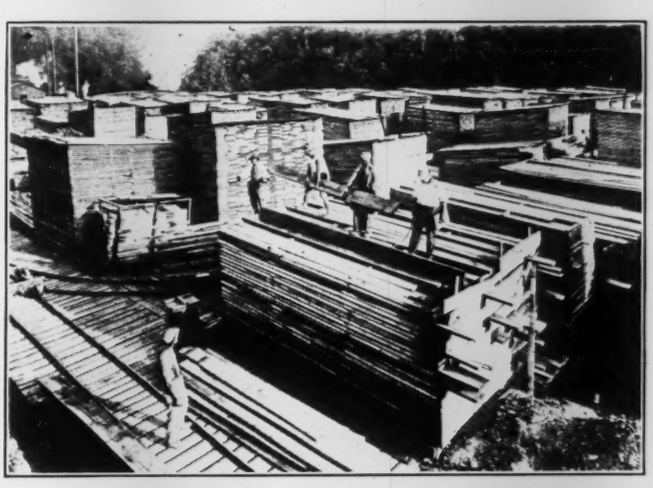

FIG. 23 UNSTACKING LUMBER TO BE LOADED IN CARS FOR SHIPMENT

FIG. 24 INSPECTING LUMBER ABOUT TO BE LOADED FOR SHIPMENT

the heart of the Appalachians. In this one gets a glimpse of the sawmill building, the planing mills, dry kilns, a part of the lumber yards, a boarding house, a club house, several residences, etc. On account of the tortuous course of the narrow valley in which the plant is located, the offices, stores, church, school building, and most of the residences are not visible.

The Design of Motor-Bus Bodies

A Discussion of the Various Points Considered by Bus Purchasers, Such as Utility, Appearance, Comfort, Strength, Lightness, Durability, Ease of Repair, Etc.—The Problem of Changing Styles—Methods of Design—Materials Available, Etc.

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THERE is a certain similarity between a motor-bus body and a house so far as the relation to the woodworking industry is concerned. A house may be built of wood, or steel framing, with many varieties of covering and many styles of architecture.

FIG. 1 AN EARLY TYPE OF BUS

Satisfactory bus bodies have been built entirely of metal. On the other hand, it is very doubtful if the all-metal bus body will ever come into general use for a long time, for reasons which will be apparent on going farther into the subject. The problem of designing a bus body is tied up with a great many factors other than the mere matter of simply handling wood.

The company with which the author is associated, and most other companies in the motor-bus business, are generally considered as members of the metal-working industry. They are accustomed, when they have a new problem to solve, to think of it in terms of castings, forgings, pressings, screw parts, etc., rather than turn to wood. Other things being equal, they prefer to use a metal body instead of one of wood. The importance of this point is the realization that wherever wood has been used in bus-body construction it is because it has been found to be the most suitable material for the purpose and not because it happened to be the easiest thing at hand. It is unnecessary to bring out in detail the points of superiority of wood for these uses, but mention may be made of three or four.

First is the matter of interior finish attainable; second is the ease of shaping and fitting; third is the ability to absorb shock and vibration without transmitting it to adjoining members; fourth is the ease of attachment of parts.

On the other hand, drawbacks to wood as a material for use in bus bodies are: first, the trouble with swelling and shrinking due to weather conditions; second, deterioration of wood due to rot; third, inability to hold lacquer satisfactorily; fourth, high first cost of the material, which may not be compensated for by its ease of working.

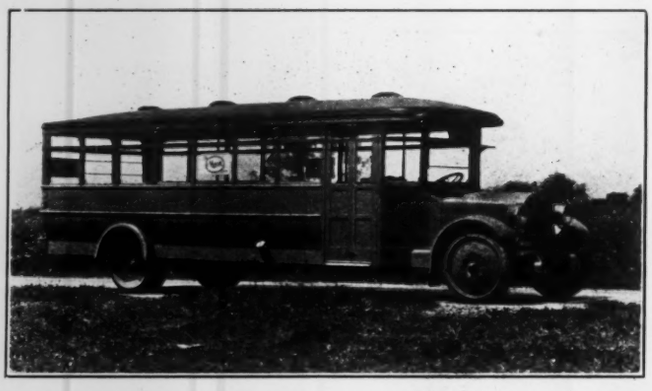

FIG. 2 WHAT A FEW YEARS' EVOLUTION HAS DONE TO FIG. 1

The foregoing are a few of the obvious advantages and disadvantages of the use of wood in bus bodies, and the subject of design will now be taken up in more detail, bringing out some of the problems and their solutions.

Going back to the similarity between a bus and a house, consider now the customer who goes out today to buy a bus. In the first place, there is no average bus user any more than there is an average house hunter. The bus customer may be the operating manager of a large public-service company who has had years of experience in public transportation work and who has some very definite ideas as to just what his requirements are, or he may be any one just starting up with a small fleet and without extensive experience, whose ideas of bus requirements may not be based on solid ground.

It is possible, however, even considering these widely different types of bus customers, to say that both are looking for two things and that the only difference between these customers is one of degree. The two things they are looking for are, first, visible quality, and second, invisible merit.

VISIBLE QUALITY

The first point for consideration under the heading of visible quality is the matter of utility. The general evolution of bus-body designs has produced several types particularly adapted for certain kinds of work. One of the variables is the matter of carrying capacity. Twenty-nine or thirty-three passengers seems to be about the limit in seating space available in single-deck bodies, keeping within the various state laws as to length and width. Buses for 16, 19, and 21 passengers have been built for service too meager to support the larger vehicles. Double-deck buses holding 65 passengers are built to handle crowded city service. The arrangement and type of seats is of course tied up with this matter of utility. For crowded city work less comfortable seats are used, and more seats can be placed in the same space. The arrangement of entrances and exits is governed by the type of service

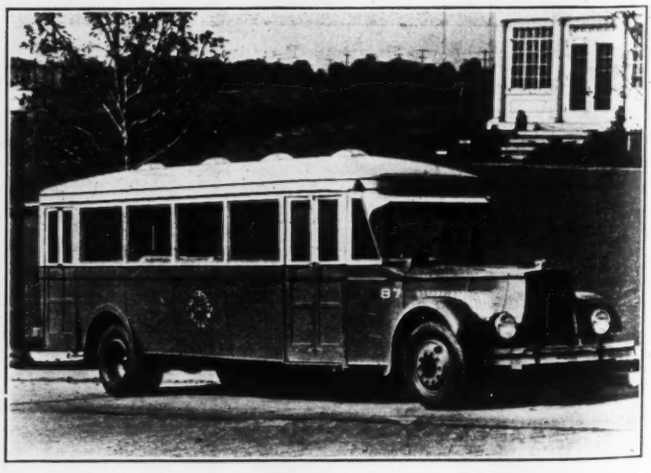

FIG. 3 A VERY RECENT CITY-TYPE BUS

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required, as well as by state laws in the case of emergency exits. These matters are gradually becoming standardized. The question of headroom is also dependent on utility. A city bus built to carry some standing load requires about 75 in. headroom, whereas the de luxe bus for cross-country long-distance work can be built with headroom down to 62 or 63 in. In designing a body to meet the customer's requirements these matters of utility are carefully considered, and it is always possible to give the customer exactly the type of body he desires.

The point of importance next to utility is the matter of appearance. From its very definition appearance means the effect of the image of the bus on some one's eye. The question is whose eye is to be affected. The novice bus buyer considers his own eye as the judge of appearance. The experienced bus buyer considers the eyes of his would-be customer riders. It is a well-known fact and has been demonstrated many times where competing bus lines are in operation that people will wait on the street corner to ride in a certain make of bus because they prefer its appearance either outside or inside. Appearance therefore covers everything there is about a bus to look at—the color of the paint, the lines of the body, the interior decorations, and above all the condition in which the operator keeps his equipment. The motor bus is being operated in competitive work either against other motor buses or against other methods of transportation. One of the

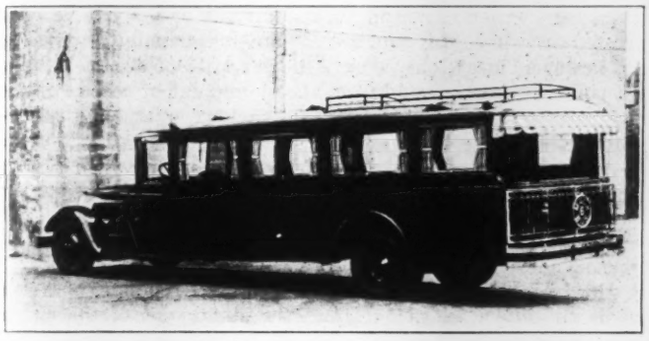

FIG. 4 THE PARLOR OBSERVATION TYPE

FIG. 5 INTERIOR OF BUS FOR SUBURBAN SERVICE

reasons for the success of the motor bus as against other forms of transportation has been its ability to attract riders away from the latter, even at a higher charge per mile. Appearance is one of the most important items that attracts riders. The matter of utility, as considered above, does not have a great effect on attracting riders. For example, take the case of a company operating de luxe type parlor buses for city service. There is no question but what the attractiveness of this bus attracts riders, but the buses are not at all convenient or well adapted to the type of service that is being given. In this case it would be possible to attract just as many riders with a well-designed city-type bus, and the operator would be giving better service at a greater profit. This matter of appearance is one of the hardest problems to meet because it covers such matters as choice of details, most of which are governed by styles or personal preferences; just as in picking out a house one person prefers a certain style of architecture, color scheme, and interior decorations, whereas this same arrangement would appear hideous to another person. The gradual evolution of railroad-car architecture from early Pullman with its ornate inlays, filigrees, and extreme ornamentation to the very quiet one- or two-tone simplicity of the present day is an illustration. In general the tendency in bus appearance seems to be along similar lines of simplicity. This is right, from the standpoint that sim-

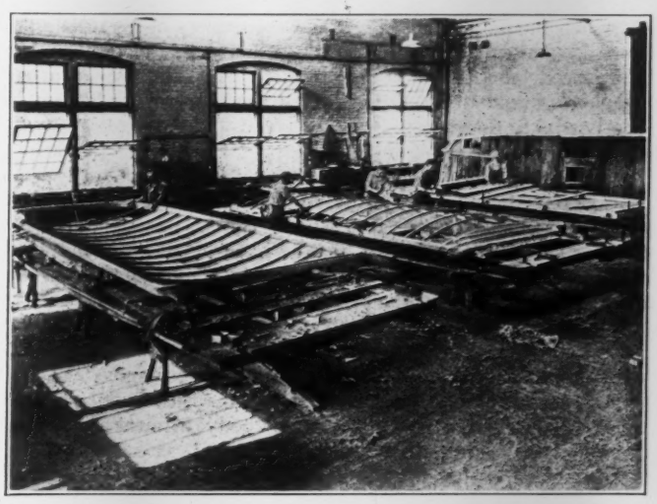

FIG. 6 ROOF SUB-ASSEMBLIES

plicity can never be offensive, whereas over-ornamentation is apt to please only a very few.

Next to appearance comes the matter of comfort to patrons. The points which should be considered under comfort do not affect in detail the design of the bus body so far as framing, etc., is concerned, but are associated with the general construction of the body design as a whole. The important point in designing for rider comfort is that there must be no effects that are annoying to any of the senses. The length of ride is connected with this matter of comfort. If the passenger is to ride all day one must go to great pains to promote comfort, whereas if a passenger is to ride only fifteen minutes the subject is less important and attention can be centered more on some of the other factors in design. The various points brought out by consideration of comfort are more or less obvious and will be considered briefly.

From the standpoint of the eyes, windows should afford a satisfactory view; post spacings should not be too close or the tops of the windows too low, causing people to lean over to see out. Also, for night driving an adequate lighting arrangement without glare should be provided. Considering the ears, all squeaks and rattles should be removed from the body, as well as drumming sounds or noises coming from the chassis. The commonest of these are noises from the engine and transmission and from the muffler. Adequate ventilation must be provided at the rate of approximately 350 cu. ft. of new air per hour per person. This will prevent headache and drowsiness on a long ride.

Temperature is a matter of importance. A heating system should be provided, preferably in conjunction with the ventilating system. On this point is based one of the most serious defects in all metal bodies. Wood has heat-insulating properties which a metal body does not have and it is necessary to use heat insulation with a metal body, which increases weight and cost. The choice of seats is important in considering comfort, and the type chosen depends somewhat on the question of utility. For a long ride a larger seat can be used at some sacrifice of carrying capacity. Seats, however, cannot provide entire riding comfort. The design of chassis and springs is the most important part of this problem. Good seats will help but cannot alone make a good-riding bus. Another sense which is important in promoting

comfort is the sense of touch from the standpoint of vibration. There are several kinds of vibration produced in a motor bus. There is the vibration coming from the road and springs, which is a problem of chassis design. There is the vibration coming from the engine and power equipment, which is a problem both of chassis and body design. Even the most perfect design of engine and transmission is usually dependent upon metal-to-metal contacts, and the vibrations must be damped out at some point before they reach the passenger in the body.

These three factors of utility, appearance, and comfort are the things that the customer looks for in buying a bus. They are the things that he can easily evaluate by walking around it and taking a short demonstration ride. They are also the three factors upon which the patron or rider judges the quality of his ride. The patron also judges his ride by certain other factors of performance such as speed, smoothness of acceleration, arrival on time, courtesy of employees, and other things which are outside of the control of the bus-body designer.

Invisible Merit

This covers those points in body construction and design which the bus customer expects but which he cannot be sure that he is getting. They are the items which he has to take on faith, either based on the past performance of similar vehicles made by the same manufacturer or on the manufacturer's reputation for quality workmanship. The most important one is that of strength, or safety, and when strength is mentioned, lightness should come in the same breath. These two factors are not opposites. The first aim should be to secure strength—in other words, proper proportions of all the parts of the bus body to meet the service requirements. Lightness, on the other hand, means no unnecessary weight and that the mechanics of the design should be such as to use the full strength of all the parts. In other words, aim for strength and secure lightness by excellence of design. Some of the problems introduced by this requirement are very difficult. It is comparatively easy for an engineer to design a bridge to span a certain distance and carry a certain load, but even in spite of the ease of his problem he multiplies his result by a factor of safety of four or five to take care of contingencies. In designing a bus body there is no such exact information on which to proceed.

At this point it may be said that there has never been developed any simple method of stress analysis in bus bodies. Photographs of buses that have been in wrecks, collisions, etc. have been collected and have determined certain points of weakness. Buses that have been in service many years have been examined and have determined the points of deterioration, and from this general collection of data perfected designs are arrived at. Most engineers like to reduce these matters of strength to a system of formulas. It is doubtful, however, if any such results will ever be arrived at in the design of bus bodies. The loads that are applied to the bodies are impossible to analyze and the structure is so complicated and of such composite design, using several materials, that it is impossible to work out any stress-distribution diagrams. Suffice it to say at the present time that one of the things a bus customer is looking for but which he cannot see is strength and lightness. He will not find out whether he has obtained these qualities until he has had the bus in service for several years.

Another point which is closely allied with strength but which deserves separate treatment is the matter of security against deterioration. A bus body must not go to pieces before its useful lifetime, either from the loosening of joints or from the weathering of wood and metal. As far as metal parts are concerned, this means proper protection of hidden surfaces during assembly and proper securing of tight joints by riveting or bolting. From the standpoint of wood parts, the requirement is met by the proper selection of varieties of wood for use in the body, the curing of this wood, and its protection against weathering with paint or other preservative. The best practice is to cover in all exterior exposed wood with a metal sheet, a light-gage aluminum being used for this purpose. The aluminum not only takes paint better than the wood, but serves as a perfect protection for exterior woodwork against weathering. The matter of fastening joints against looseness is important. Some of the earlier buses were put together with screws at the more important points and nails at the less important. Joints between wood were mortised, glued, or white-leaded, and doweled. As experience has been gained these requirements have been tightened up. Where screws were used before, through bolts with nuts are now used. In many places where nails were employed before, screws are now used—for example, in fastening down the roof, putting on moldings, etc. Joints have been improved both by the quality of workmanship and strict control of gluing as well as by the use of steel connecting angles and gussets wherever possible so as to prevent the working and loosening of joints.

One last item which the customer buys but cannot discover until he has had the bus in service is the matter of ease of repair. It is impossible to predict just what accidents a bus body is likely to encounter. If this were possible, bodies could be specially designed for the insertion of repair parts. All sorts of accidents, however, occur to bus bodies, ranging from a small dent in a side panel and scratches and mars on the interior finish, to the smashing in of a main post or sill in a collision. There are many arguments pro and con for wood or steel from the standpoint of ease of repair. A structure built entirely of wood can be very easily repaired by a skilful carpenter with the aid of a few tools and a pot of glue. On the other hand, a metal body is more difficult to repair, and unless entirely new parts are used it is likely to carry some scars. One of the commonest types of accident is the injuring of the side panels. For this reason several designs of body have been suggested in which the side panels are individually removable. This, however, meets only one particular kind of accident. The main point to keep in mind from the standpoint of ease of repair is that it should not be necessary to take down a large part of the inside trimming of a bus and remove seats, etc., in order to make some minor repair to the outside, just as it should not be necessary to remove the radiator and fan of an engine in order to put a new packing in the water pump.

Change in Styles

A third consideration which comes into the design of a bus body is entirely a selfish one from the standpoint of the manufacturer and covers the points that he must pay attention to in order to make a successful bus which can be sold at a profit. The most important of these is the matter of meeting the change in styles. There is certainly no such thing as standardization in the manufacture of buses at the present day. In the first place, there is too much competition. In the second place, the operators of buses have not been in the business long enough to have determined the possibilities of their use and therefore the types of design best fitted for their needs. It would be possible to standardize on a certain type of bus body for manufacture in quantities and the cost of these bodies would undoubtedly be much less than at present, but it is very doubtful if they could be sold in quantities large enough to warrant making them. It is most important, therefore, that the bus manufacturer today be prepared to meet every possible change in style. In the severe competition going on the manufacturer who cannot easily meet these changes will be left behind with obsolete models.

This question of standardized design, however, is largely one of degree. Most bus customers expect deliveries a few weeks from the time of placing their order. Obviously it would not be possible to start on a blank sheet of paper to design a body, build it, and deliver it in this short space of time. It is necessary, therefore, for the bus-body manufacturer to have available and in various degrees of manufacture certain standard designs of component parts of buses which can be worked up into a number of different complete bodies that will meet the majority of requirements. The customer, then, who wants a complete custom body absolutely different from every one else's must, of course, wait for the whole process of development, but the average customer who is in a hurry to get delivery is willing to take this semi-standardized body. Presumably as the use of buses becomes older the amount of standardization possible will increase, but it is doubtful if it will ever be possible to finish up complete buses in large quantities and have them on the showroom floor ready for delivery in the same way that pleasure cars are sold.

This matter of ease of change of styles is one of the worries

of the bus manufacturer. To manufacture buses conveniently it is necessary to use jigs for setting up the bodies, but when the bodies are built on the basis of ten being of one style and the next fifteen of another, it is necessary that these jigs be made adaptable to various combinations such as length, width, post spacings, arrangement of doors, height of roof, curvature of roof, shape of cowl, etc. Change of styles is the principal reason why the all-metal bus body is not likely to come into use in the near future. A metal bus body built up out of standard rolled shapes and sheets is very heavy and clumsy in design. A satisfactory metal bus body can be built only by the use of specially designed pressed-steel parts. The investment in dies and tools to make these parts for one design of bus body might amount to $50,000. It is obvious that there would be no profit in this type of body unless a very large number all alike could be sold. On the other hand, a design made up of wood frame but with metal connecting pieces can be worked out so that by very simple changes in the jigs, bodies of almost any style can be built.

The second point is the matter of ease of manufacture. It is really very much the same factor under another heading. Consider, for example, the common ball corner that was introduced at the upper rear corner of various closed cars several years ago. This type of corner is an illustration of one of the problems of bus design. If there are many thousands of these corners to be made, there is no problem at all. It is simply a matter of a set of dies costing possibly two thousand dollars, and pressing these corners exactly to shape out of sheet metal. If, however, there are only a few to be made, the die cost is unreasonable. If they are to be made of sheet metal, the only method is to take a flat sheet, cut some gores, weld the edges, and then hammer the sheet out on a power hammer until it is bent to shape. The welding operation is done by a highly skilled man with the use of expensive equipment, and the hammering operation is probably one of the most expensive, both in wages and equipment, that can be found in any sheet-metal shop. One man and a helper could probably turn out about three of these ball corners a day. On the other hand, consider wood in this design. The process is the simple one of gluing up a large block of wood, sawing it out roughly on a band saw, and then shaving it off to meet certain lines of a templet. This wooden block can then be covered with black fabric, furnishing a perfectly good-looking corner at a fraction of the cost of a metal corner.

The next important point in this question of manufacturing for profit is of course the matter of cost. Whether it is called readiness for change of styles or ease of manufacturing, the underlying question is one of cost to produce a certain quality of design. If the costs are to be low, bodies must be manufactured and not just built. The small job or custom shops build their bodies. They start out with a stand on which are set up some sills and posts, and the framing is worked out as they go along. The body builder measures each piece and fits it into place. On the other hand, to manufacture bodies the work must be divided. Detail parts must be made up in advance in quantities by regular machining methods. These are brought together in sub-assemblies and these, in turn, gathered into the final assembly on the body jig. The sub-assembly and the final assembly are both made on jigs. With these factors in mind it is possible to design a series of sub-assemblies which with very little variation can be gathered together in different combinations to form the different styles of bodies required. Parts that go into these sub-assemblies can be made either in a metal-working shop or in a woodworking shop by well-known machine methods from shaper patterns, drill jigs, and templets.

METHODS OF DESIGN

One of the first problems in designing a body is the question of its suitability to the chassis on which it is to be mounted. Chassis design will be only mentioned at this point to bring out one point, namely, the question of a flexible or a rigid chassis. The terms are relative ones as there is no completely rigid chassis. A bus chassis may have a wheelbase of from 15 to 20 ft. and be operated at high speed over uneven ground. In accordance with the best principles of chassis design, the connecting members between the front and rear axles must allow for considerable flexibility so that the wheels can follow the ground over any roughness without depending too much upon the springs to do this work. This means a flexible frame and requires that the various units mounted in the frame such as engine, transmission, etc., shall either be mounted on three-point suspension or with some sort of flexible mounting such as rubber. A truly rigid frame, on the other hand, would not yield to the unevenness of the ground and would require that it all be taken care of in the springs. This would result in considerable pounding and hard riding. Moreover a rigid frame, to be truly rigid, would be impossible as to weight. Loads on buses of the above size run as high as 25,000 lb. The impact values in high-speed operation might multiply this static load by four. A rigid beam to meet this would possibly weigh as much as 2500 to 3000 lb. for a length of 20 ft. between supports. This weight would be prohibitive. It is desirable also not to depend on the springs too much for flexibility as the weight is thus taken off the tires, allowing slipping and skidding.

Considering, then, that a flexible chassis will be used, what kind of body should be put on the frame? As explained above, this question of flexibility is one of degree, but in order to define what is meant by flexibility it may be said that a bus chassis frame with a 20-ft. wheelbase should be capable of allowing one wheel to drop into a 6-in. depression without any appreciable reduction in the load on the springs. Moreover the same frame, when loaded with, say, 20,000 lb., will have a total deflection of $\frac{1}{4}$ to $\frac{3}{8}$ in. at the center.

It would be entirely possible to make a body of flexible design which could be mounted on this flexible chassis. The so-called Weyman bodies that are built in Europe follow this idea. They are more in the nature of a tent, consisting of frame members with yielding connections and covered with some sort of fabric. This type of body has not so far been popular in America and has not yet demonstrated its suitability for bus work, though it may be desirable for small-size pleasure cars. The disadvantages of this type are, first, the probable objection of the riding public to sitting in a structure in which movement can be noted in all the members; second, the weakness of the structure in collision or in rolling over in a ditch; and third, the probable depreciation of the structure in service over some of the roads to be found in this country.

The demand in this country appears to be for a rigid body. From experience with the wear and tear and depreciation on bodies, the author would say that the stiffer a body can be made the longer it will last—and this means many times more rigid than the chassis. It is found, for instance, from tests that the body is six times as stiff in vertical bending as the chassis frame on which it is mounted, although the loads imposed on the body, and particularly the impact loads, are very much less than those imposed on the chassis frame. This means, therefore, that there must be an allowance in the method of attachment of body to chassis to take care of motion. This is done in several ways. First, by mounting the body on some sort of fabric resting on the two main chassis rails which are relatively close to the center line, and balancing the outside of the body on outriggers attached to the rails, these outriggers being made of relatively light material which will allow for some motion. With these closely spaced center supports it is possible for the chassis frame to twist considerably without affecting the body, and the fabric strips between the body and chassis permit a certain amount of deflection. Another method which carries the same idea farther is to mount the body with a sort of three-point suspension, the rear of the body attached as above to the chassis frame, while the front, which is in line with the point of maximum deflection in the chassis frame, is carried in rubber shock-insulator blocks attached to suitable outriggers on the frame. In this way the rear of the body over the rear wheels is fastened rigidly to the chassis, whereas the front of the body is free to move with respect to the chassis within the limits of the rubber shock insulators. This appears to be a very desirable arrangement.

The question of designing a stiff body brings up several points. The first buses built followed somewhat the lines of steam-railroad practice. A floor was constructed with heavy sills of wood or steel, and on this a superstructure was erected. The superstructure was relatively light and designed more or less to support the roof and side walls, but did not contribute particularly to the strength

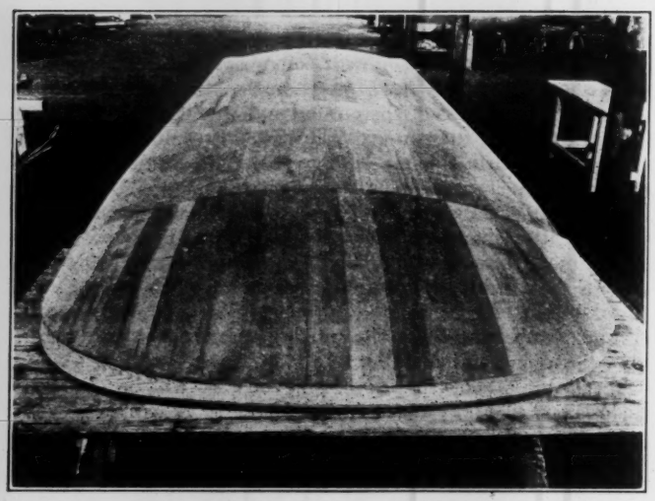

FIG. 7 A FINISHED ROOF ASSEMBLY

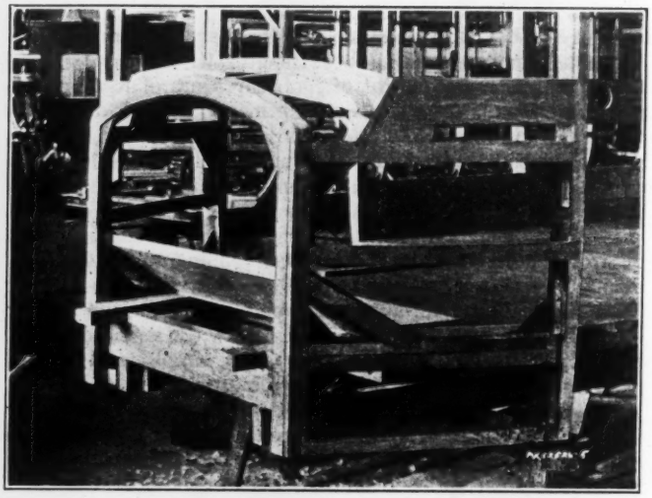

FIG. 8 A COWL SUB-ASSEMBLY

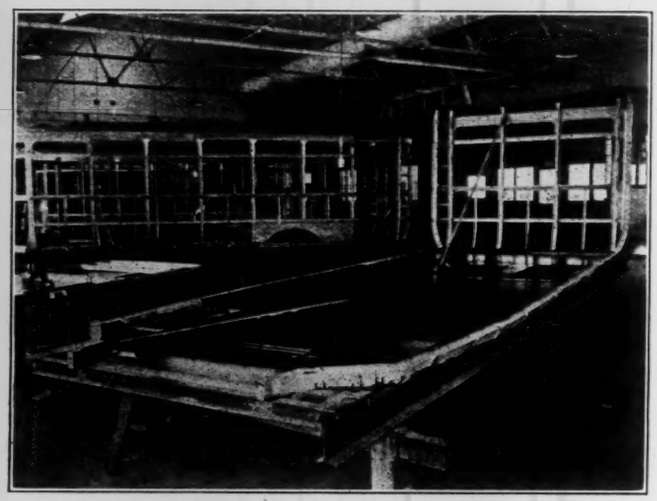

FIG. 9 COMMENCING THE FINAL FRAME ASSEMBLY

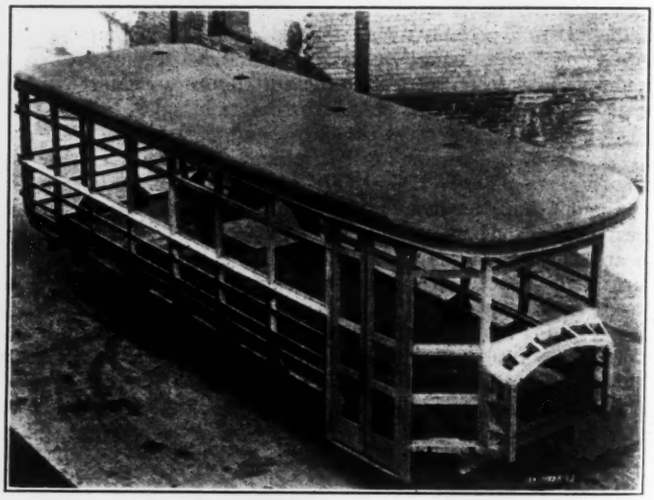

FIG. 10 EXTERIOR OF COMPLETELY FRAMED BODY

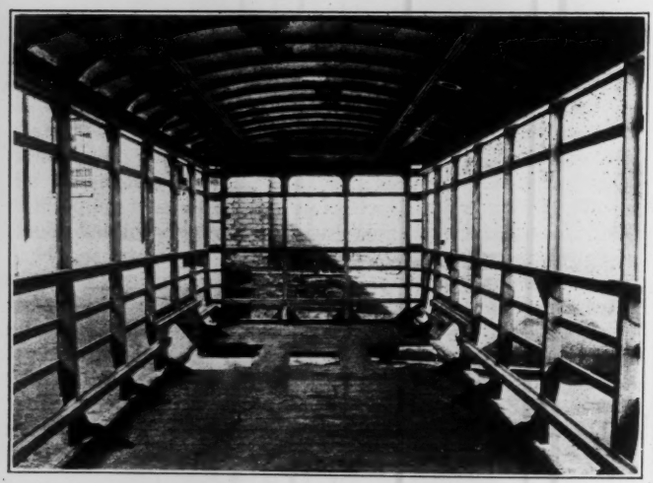

FIG. 11 INTERIOR OF BODY FRAMING

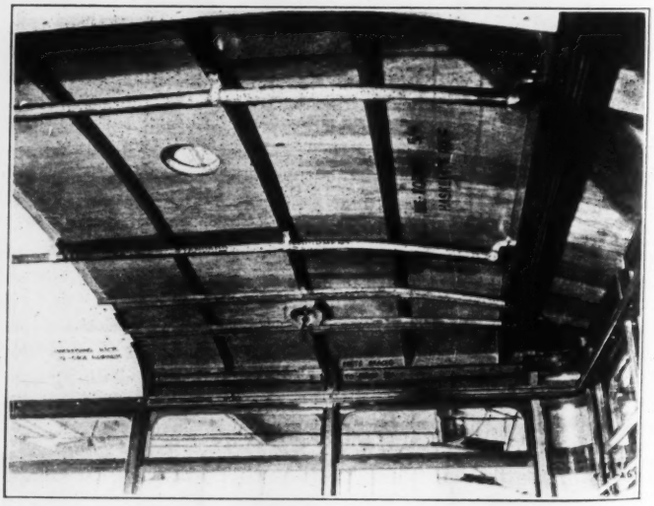

FIG. 12 EXPOSED VIEW INSIDE OF ROOF ON COMPLETED BUS

of the body. The later theories on bus-body design are in accord with the very latest theory on light-weight steel railroad-car design in which the body is considered as a complete closed box and every part of the structure is made to contribute something toward the strength of the whole. If a body were simply a tube with both ends closed it would have the strongest possible structure for a given weight to withstand the bending and torsion, which are the two enemies of bus-body design. Of these two, torsion is by far the more serious.

Obviously one cannot design bus bodies to look like a tank car, but can follow this general theory of a flat-sided, closed box having a length somewhat greater than the other dimensions. Consider such a box and note that when a bending load is imposed the top is in compression, the bottom is in tension, the two ends are not disturbed, and that the sides, being relatively thin, undergo a buckling force, partly tension and partly compression. Note further that if a torsion load is put on this type of box every one of the six surfaces resists a distorting stress, and that, conversely,

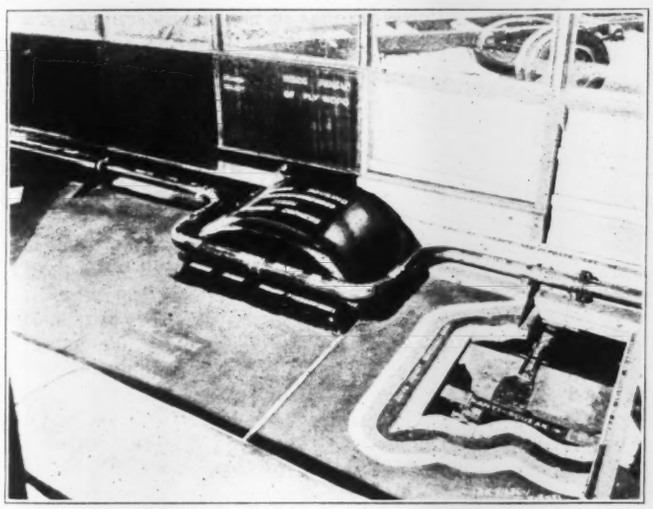

FIG. 13 ARRANGEMENT OF WHEEL HOUSING AND HEATER PIPES. FLOOR CUT AWAY TO SHOW OUTRIGGER. ATTACHMENT OF INSIDE PANELS

any stiffness put into any of the corners of the structure will help to resist this load. This is the important point to keep in mind in designing a bus body along this theory. Start with a closed box of equal strength throughout. It then becomes necessary to cut some large holes such as for the front end around the windshield, etc. and for the doors. Reinforce around these holes until the original strength in that particular surface has been regained. This can only be done by the liberal use of steel gussets and angles. The openings for windows, etc. are not large enough to be a serious problem for bracing. On the other hand, any large surfaces must be watched to see that they are properly supported against crumpling.

Following these general principles, there is no serious difficulty in designing a rigid body. The only additional problem is that of designing to meet certain special problems that come up at certain points. For instance, a common form of bus collision is the case of "side swiping," and the joints between the post and side sills should therefore be strong enough to stand quite an impact at this point. Another important point is the matter of designing around the doors from the standpoint of the continual blows struck when the doors are slammed shut. The same is true as regards the bracing for mounting such special heavy parts as tire carriers, etc.

Coming now to the question of the various materials available for use in body building, as was said at the outset, bus bodies cannot be built of any one material. There have been so-called all-steel bodies, all-aluminum bodies, and all-wood bodies, but the best body today is one in which the designer has no particular ax to grind in regard to any one material. Wood is certainly the most desirable material for a large part of the framing, as brought out above. Of the various woods, a high-grade heavy ash seems to be the best for framework. Some body builders use maple for this purpose, but maple will not stand the weather and is apt to rot out if damp. Ash, pine, cypress, and poplar are used for flooring. Poplar is frequently used for small, unimportant parts in the body construction. Mahogany or walnut, either solid or in the form of veneering, is used for the interior trim. Haskelite is frequently used for roofs, and sometimes Plymetal, whereas other builders build roofs more like those of a trolley car, which is a construction of poplar slats covered with heavy duck and heavily painted.

The body builder who works on a large scale has a real problem in his lumber requirements. The ordinary-size bus body may use from 1000 to 1200 ft. of unfinished lumber per body. It is important to have a very close tie-in between the purchasing of this lumber and the designing of the body as regards sizes and kinds of timber available. If any kind of a result is desired in the finished structure, this lumber must be very carefully kiln-dried. The problem of cutting the lumber and working it up is not different from that in any woodworking industry; the same general principles of laying out, machining, and assembly are followed.

Probably the next most important material in the building of bus bodies is aluminum. Aluminum is used in the form of sheets, tubing, and small castings. The sheets are mostly of the half-hard finish, which is easily worked yet not too easily dented or marred. It is the practice with some body builders to buy this aluminum in the bright finish, thereby being able to detect more easily any waves or imperfections in the finished body. Other body builders buy it in the dull scratch-brush finish. In any case it is necessary that this aluminum be all scratch-brushed

FIG. 14 EXPOSED VIEW OF REAR INSIDE

before the painting operation. Some body builders are using Plymetal for large outside sheets and still others are using steel. There are some slight differences between steel and aluminum for this purpose. The aluminum will probably weigh less on the whole is a little easier to work, and when properly handled will hold paint better than the steel. On the other hand, it costs more. Aluminum tubing is used for stanchions, hand rails, and such places, although some body builders use steel tubing for these purposes and in some cases a baked enamel finish is put on. Numerous aluminum castings are used for various minor pieces around the bodies, as well as in seat construction.

Steel, malleable iron, and gray iron are used in a few places around the body. A great deal of steel is used in the form of pressed shapes for gussets and reinforcing attachments. Sometimes malleable castings are used for these purposes and also in connection with such equipment as door-operating mechanisms, seats, etc.

One of the things that comes into bus-body designing and building and which covers quite a field is the large amount of leather and fabric that is used for seat coverings, curtains, and interior finish. Some of the very finest and highest-priced genuine leathers are used in modern de luxe buses. It may be that this is merely a passing phase, as it is doubtful whether the ultimate riding public

demands a seat covering or interior finish of a finer grade than will be found in the most expensive homes and clubs. The same is true of silk curtains, etc. The tendency appears to be toward the use of more of the lower-priced leathers for seats, of imitation leather for interior finish, and of simple shades such as used in street cars instead of the more expensive draperies. On the other hand, this question of luxury versus utility is an important one. The growth of the motor bus has been due largely to two things: its ability to furnish transportation under conditions where other forms of transportation could not thrive, and its ability to take transportation away from some of the existing forms by furnishing a more attractive service for one reason or another. One of the things that has attracted people to the cross-country motor bus as against the steam train or the interurban trolley has been this very matter of luxuriousness of appointments. It may appear uneconomic to see a farmer covered with mud or a mechanic in greasy overalls riding in a bus finished with appointments equal to those of the best metropolitan club, and yet possibly the very fact that he does not have any other opportunity to taste this luxury may be the reason that he rides on the bus in preference to the steam railway or trolley car.

On the other hand, if a bus operator has this kind of bus equipment he has a real expense to meet in keeping it up, as there is nothing so forlorn looking as a de luxe bus which has been allowed to run down at the heel.

A number of materials are also used for floor coverings. The majority of bus floors are covered with linoleum, although some of the cheaper buses are using plain wood floors with slats of some kind. The higher priced de luxe buses frequently use tiled rubber. This is very heavy, high in price, and has no particular advantage over linoleum as regards wear. A few buses are being equipped with composition flooring similar to steam railroad cars. This has not been proved entirely successful, although it appears to have some merit.

Specific Characteristics for Hydraulic Turbines

A Suggested New Expression for Specific Speed That Is Non-Dimensional and Embraces the Three Fundamental Characteristics of a Turbine: Namely, Peripheral-Speed Coefficient, Velocity-of-Flow Coefficient, and Efficiency

By ARNOLD PFAU,[1] MILWAUKEE, WIS.

THE objection raised by Bruno Eck in his article in the September, 1926, issue of MECHANICAL ENGINEERING,[2] namely, that the characteristic (or specific) speed is not non-dimensional because it is based upon the units of the respective dimensional system, is justified.

The specific speed of any hydraulic turbine (action, reaction, propeller, or impulse type) is the speed in revolutions per minute of a model so proportioned that it develops unit horsepower under unit head. The unit of head can be expressed in the metric, English, or any other dimensional system. Therefore each system has its own absolute value of specific speed n_s.

$$n_s = \frac{n\sqrt{N}}{H^{5/4}}$$

where n_s = specific speed
N = horsepower
H = unit of head
n = revolutions per minute.

n is an absolute numerical value. N is a conventionally adopted and practically fixed rate at which work is performed and is equal to 75 kg-m. per sec. (735.4 watts) in the metric system and to 550 ft-lb. per sec. (745.7 watts) in the English system.

The unit of measurement of power is the horsepower. Unfortunately this unit is not of equal absolute value for the various systems. For instance, the metric horsepower is equal to 735.4 watts (or 542.4 ft-lb. per sec.), while the English horsepower is 745.7 watts (or 550 ft-lb. per sec.), the unit of measuring electric energy being the watt and, fortunately, the same in any system.

The metric horsepower is smaller than the English, consequently to develop unit horsepower in the metric system the turbine runner can be a trifle smaller and therefore runs a trifle faster; and thus the specific speed n_s appears a trifle higher than it is when calculated on the basis of the English horsepower. This can be duly rectified by introducing an additional transformation factor K.

$$N_1 = K_1 W, \quad \text{or} \quad \frac{N_1}{W} = K_1$$

Therefore the values of n_s can be transformed from one system to another by multiplying them by a transformation factor, as follows. If the horsepower N were the same absolute value the transformation factor would be $m^{5/4}$, in which m expresses a linear measurement.

1 ft. = 0.305 meter, or $m_1 = 0.305$
1 m. = 3.28 ft., or $m_2 = 3.28$

Thus $(3.28)^{5/4} = 4.414$, and considering the difference in the absolute values of the horsepower, we have to multiply by the ratio K_1/K_2 between the measures of horsepower in the two respective systems. For instance, from English to metric:

$$m = \frac{1}{0.305} = 3.28$$

$$m^{5/4} = 4.414$$

$$\frac{K_1}{K_2} = \frac{550}{542.4} = \frac{745.7}{735.4}$$

and

$$\sqrt{\frac{K_1}{K_2}} = 1.0069$$

Thus the correct transformation factor from the English to the metric system is

$$m^{5/4} \times \left(\frac{K_1}{K_2}\right)^{1/2} = 4.414 \times 1.0069$$
$$= \underline{4.454}$$

The author would suggest here that the dimensional value n_s of the specific speed be modified by dividing it by those inherent

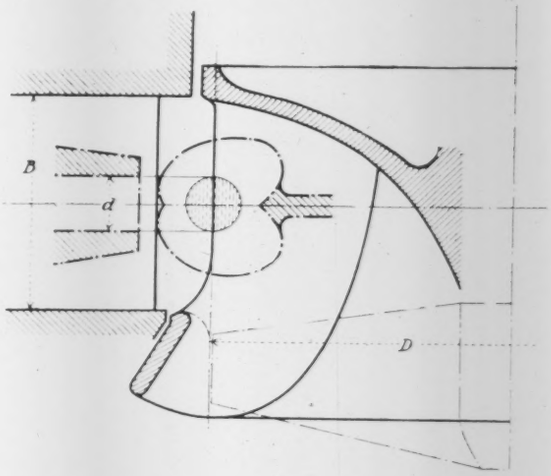

FIG. 1

values which are subject to the respective dimensional system and to its definition of horsepower. To do this it is necessary to substitute for n and N those values which contain the dimensional and conventional factors:

$$n = \frac{60\, u^z \sqrt{2gH}}{\pi D}$$

$$N = \frac{KQH\gamma e}{r} = \frac{K_1 \pi D^2 V^z \sqrt{2gH}\, He\gamma}{4r}$$

in which

u^z = peripheral coefficient of speed at the diameter D of runner, as shown in Fig. 1
π = 3.14159 (constant)
g = gravity (dimensional) = 9.81 m. per sec. per sec., 32.2 ft. per sec. per sec., etc.
D = diameter (dimensional) of runner for which the peripheral speed u or $u^z = u/\sqrt{2gH}$ is figured
$V^z = V/\sqrt{2gH}$ = coefficient of velocity of the quantity of water Q passing through a circular section of diameter D
γ = weight (dimensional) of one unit of volume (1000 kg. per cu. m. or 62.4 lb. per cu. ft.)
e = efficiency of turbine
r = number of units (dimensional) constituting one horsepower (550 ft-lb. per sec., 75 kg-m. per sec., etc.)
K = factor with which the horsepower of the system must be multiplied in order to reduce it to a uniform absolute basis of energy. For instance, (metric)K_1 = 0.7354; (English)K_2 = 0.7457.

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[2] Characteristic Coefficients for Hydraulic Machines. Reprinted for convenient reference at the close of the present paper.

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Be sure to bring this copy with you to the Spring Meeting

Thus substituting,

$$n_s = \frac{n\sqrt{KN}}{H^{5/4}} = \frac{60u^x\sqrt{2gH}}{\pi D} \cdot \frac{\sqrt{\pi/4}\, D\sqrt{V^x}\cdot\sqrt{2gH}\cdot\sqrt{H}}{H^{5/4}} \cdot \frac{\sqrt{Ke\gamma/r}}{4}$$

or regrouped,

$$n_s = 16.925\, u^x \sqrt{V^x}\sqrt{e} \times \sqrt{K\gamma/r}\sqrt[4]{(2g)^3}$$

The value thus obtained is composed of three groups of values, viz.:

(1) 16.925, an absolute constant.

(2) $u^x\sqrt{e}\cdot\sqrt{v^x}$, product of three characteristic coefficients determining the specific speed and independent of dimensional systems.

(3) $\sqrt{K\gamma/r}\cdot\sqrt[4]{(2g)^3}$, values which are dependent on the dimensional system and its definition of horsepower.

It is suggested that a new definition be adopted, viz.:

$$n^x{}_s = \text{Specific characteristic} = \frac{n_s}{16.925\cdot\sqrt{K\gamma/r}\cdot\sqrt[4]{(2g)^3}} = \underline{u^x\sqrt{V^x}\cdot\sqrt{e}}$$

This value is non-dimensional and embraces the three fundamental characteristics of a turbine, namely,

1 The peripheral-speed coefficient u^x, or what may be termed the "speediness" (German, *Schnellaüfigkeit*).

2 The coefficient of velocity V^x, or what is called *Schluckfähigkeit* in German, and what we may term the "flow receptivity."

3 The efficiency e of the turbine, representing the engineering quality of design.

It can be readily seen that by overspeeding a turbine the specific speed n_s increases in direct proportion to the speed n, provided the horsepower remains the same. The overspeeding, however, reduces the efficiency e, therefore the value e should be included in the expression for the specific characteristic, and it should also have been included in the present expression for specific speed.

The dimensional values $\sqrt{K\gamma/r}$ and $\sqrt[4]{(2g)^3}$ are constants for each respective dimensional system. For instance,

Metric:

γ_1 = 1000 kg. per cu. m. of water
r_1 = 75 kg-m. per sec. per hp.
K_1 = 0.7354 kw.
g_1 = 9.81 m. per sec. per sec. (gravity)

$$\text{Thus } \sqrt{\frac{K_1\gamma_1}{r_1}}\cdot\sqrt[4]{(2g_1)^3} = \sqrt{\frac{0.7354\times 1000}{75}}\cdot\sqrt[4]{(19.62)^3} = \underline{29.191}$$

English:

γ_2 = 62.4 lb. per cu. ft. of water
r_2 = 550 ft-lb. per sec. per hp. (English)
g_2 = 32.2 ft. per sec. per sec. (gravity)
K_2 = 0.7457 kw.

$$\text{Thus } \sqrt{\frac{K_2\gamma_2}{r_2}}\cdot\sqrt[4]{(2g_2)^3} = \sqrt{\frac{0.7457\times 62.4}{550}}\cdot\sqrt[4]{(64.4)^3} = \underline{6.612}$$

Therefore the ratio between the two systems is

$$\frac{29.191}{6.612} = 4.414 = m^{5/4}$$

which is the same as previously obtained, based on dimensional terms of horsepower values.

The value $n_s = \frac{n\sqrt{N}}{H^{5/4}}$ can also be transformed in still a different way. For example, for radial-flow turbines we can express the discharge

$$Q = V_2{}^x\sqrt{2gH}\times\pi BD$$

taking the cylindrical area of the runner diameter D instead of the circular area. Here B represents the height of the guide case. B can be expressed in terms of D, viz.: $B = \alpha D$. This would change the formula for n_s by substituting for V^x the value $\alpha V_r{}^x$, in which α again represents an important design feature.

Thus

$$n_s = 16.925\cdot u^x\sqrt{e}\cdot\sqrt{\alpha V_r{}^x}\cdot\sqrt{K\gamma/r}\cdot\sqrt[4]{(2g)^3}$$

and

$$n_s{}^x = u^x\cdot\sqrt{e}\cdot\sqrt{\alpha V_r{}^x}$$

Similarly we can express the discharge Q of an impulse wheel.

$$Q = V_2{}^x\sqrt{2gH}\,\frac{\pi d^2}{4}$$

in which d represents the jet diameter, and $V_2{}^x$ the coefficient of velocity of the jet, and since the ratio of jet diameter to wheel diameter is a characteristic feature of an impulse wheel, we can set $d = \alpha_2 D$, so that $V^x = \alpha_2 V_2{}^x$

Thus

$$n_s = 16.925\cdot u^x\sqrt{e}\cdot\sqrt{\alpha_2 V_2{}^x}\sqrt{K\gamma/r}\cdot\sqrt[4]{(2g)^3}$$

all as per the general formulas, and

$$n_s{}^x = u^x\sqrt{e}\sqrt{\alpha_2 V_2{}^x}$$

The conventional term $n_s{}^x$ or "specific characteristic" would apply to any dimensional system, and its adoption would permit of its separation into the fundamental characteristics of speed, discharge, efficiency, and the principal design feature of ratio of diameter to height of guide case for radial turbines, or jet diameter for impulse wheels.

EXAMPLE

English	*Metric*
n_s = 100	n_s = 441.4

Then specific characteristic

$$n_s{}^x = \frac{100}{16.925\times 6.6123} = 0.8933 \text{ (English)}$$

or

$$\frac{441.4}{16.925\times 29.191} = 0.8933 \text{ (metric)}$$

which is now the same value in both systems.

Since the specific characteristic is the product of three ratios, or coefficients, it can be expressed as a percentage by multiplying the value defined above by 100. This would change the value given in the above example to 89.33, which is but little less than the value n_s or specific speed now used in our English dimensional system.

The Characteristic Coefficients for Hydraulic Machines

By BRUNO ECK,[1] AACHEN, GERMANY

TWO objections may be raised against the use of the characteristic (or specific) speed as a basis for comparing water turbines, centrifugal pumps, and other hydraulic machinery. These are:

1 The characteristic speed is *not* non-dimensional, which is desirable neither from the practical nor the theoretical point of view, as it is particularly troublesome to have to work with dimensional constants when comparing tests for which results are given in different systems of units, e.g., British and metric; and

2 It does not indicate the fundamental relationship that exists between turbines, centrifugal pumps, airscrews, screw propellers, windmills, etc.

The purpose of this note is to point out this fundamental rela-

[1] Aerodynamic Institute.

Reprinted from MECHANICAL ENGINEERING, September, 1926, pp. 949–950.

tionship and to derive non-dimensional coefficients related to the characteristic speed from physical considerations.

CHARACTERISTIC SPEED OF HYDRAULIC TURBINES AND CENTRIFUGAL PUMPS

"Turbine runners of different types may possess characteristics of power, speed, dimensions, and operating head which vary widely. To make available a common basis for comparison, the value designated as 'characteristic speed' is used. The characteristic speed of a runner (sometimes designated 'specific speed') is the speed in r.p.m. which a model of a runner would have if operated under a head of 1 ft., this model to be reduced proportionately in all dimensions from the original model until it will develop 1 hp. under 1 ft. head."[2]

Since the angles of the buckets of a runner are fixed, the linear speed of the bucket should at all times bear a constant relation to the linear velocity. The r.p.m. (n) of a runner is therefore proportional to $\sqrt{H}$, where H is the head in feet, and the speed at 1 ft. head ($= n/\sqrt{H} = n_1$) is called the unit speed. The power of a runner (N) is a function of the pressure and quantity of the operating water, and is therefore proportional to $H \times \sqrt{H}$, and the power of a runner at 1 ft. head is equal to $N/H\sqrt{H}$, and is called unit power (N_1).

The characteristic speed $n_s = n_1\sqrt{N_1}$

$$= \sqrt{N/H\sqrt{H}} \times \sqrt{H}$$

$$= n\sqrt{N}/H^{5/4}$$

For centrifugal pumps the formula is

$$n_s = 8.9n\sqrt{Q}/H^{3/4}$$

where Q is the weight of water discharged.

If n_s is held constant, the efficiency of the turbine or pump remains constant, or is a function only of the characteristic speed. In considering efficiency for geometrically similar machines, the Reynolds Number should theoretically be a constant. But this refinement in the application of the laws of similarity is scarcely of importance in practical engineering.

SIMILAR COEFFICIENTS FOR ALL HYDRODYNAMIC MACHINERY

It would appear that similar non-dimensional coefficients should apply to all forms of hydrodynamical machines, since the fundamental physical action is always the same: momentum is always received from solid bodies and delivered to liquids or gases, or received from liquids or gases and delivered to solid bodies.

In the theory of airscrews and of ship propellers two non-dimensional coefficients are employed:

1 Coefficient of thrust: $\sigma = \dfrac{\text{thrust}}{\text{dynamic head} \times \text{area}}$

2 Coefficient of pitch: $\lambda = \dfrac{\text{forward speed}}{\text{rotational speed}}$

We shall investigate whether analogous coefficients may be derived for turbines.

COEFFICIENTS FOR AXIAL-FLOW MACHINES

For the hydraulic turbine, then, it follows by analogy that if

g = acceleration due to gravity in ft.
γ = specific weight in lb. per cu. ft.
H = head in feet
Q = quantity of water discharged in cu. ft. per sec.
n = r.p.m.
c_m = forward speed
u = peripheral velocity, and
D = diameter of runner,

then the coefficient of thrust may be written as:

$$\sigma = \frac{\gamma H}{\frac{\gamma}{g} \times \frac{1}{2}c_m^2} = \frac{2gH}{c_m^2}$$

The following relationships also obtain:

$$c_m = \frac{Q}{\frac{\pi}{4}D^2}; \quad D = \frac{u}{n}\frac{60}{\pi}; \quad \text{and} \quad \frac{u^2}{g}\nu = H$$

(The coefficient λ will be discussed later.)

Making appropriate substitutions,

$$\sigma = \frac{\pi^2 \times g^3 \times 60^4 \times H^3}{\nu^2 \times 8Q^2 \times n^4 \times \pi^4} = \frac{CH^3}{\nu^2 Q^2 n^4}$$

where C is a constant.

Taking the characteristic-speed equation of the centrifugal pump

$$n_s = 8.9\frac{\sqrt{Q}}{H^{3/4}}$$

and substituting, we find

$$\sigma\nu^2 = \frac{240 \times 10^{11}}{n_s^4}$$

The characteristic speed is thus seen to be inversely proportional to the fourth root of $\sigma\nu^2$.

COEFFICIENTS FOR RADIAL RUNNERS

In similar fashion it is possible to introduce a coefficient of thrust for radial runners. In Fig. 1 we have a radial wheel and an axial runner of the same outside diameter. If the two discharge the same weight with the same head,

$$c_m\pi bD = c_{m1}\frac{\pi}{4}D^2$$

and

$$c_{m1} = 4\frac{b}{D}c_m$$

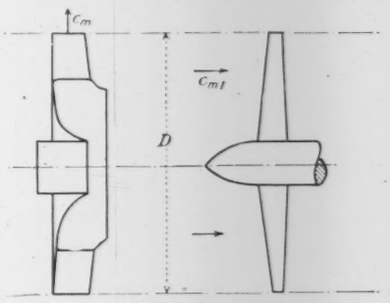

FIG. 1

These two runners have the same coefficient of thrust, and we have only to make appropriate substitutions for the radial runner to obtain

$$\sigma = \frac{\gamma H}{\frac{\gamma}{2g}c_{m1}^2} = \frac{\gamma H}{\frac{\gamma}{2g}16\left(\frac{b}{D}\right)^2 c_m^2} = \frac{D^2 H}{8b^2 c_m^2}$$

Writing

$$c_m b\pi D = Q; \quad D = \frac{u}{n}\frac{60}{\pi}; \quad \text{and} \quad \frac{u^2}{g}\nu = H,$$

gives

$$\sigma = \frac{\pi^2 g^3 60^4 H^3}{\gamma^2 8Q^2 n^4 \pi^4} = \frac{240 \times 10^{11}}{\gamma^2 n_s^4}$$

That is, the coefficient of thrust is of identical form and the characteristic speed is again inversely proportional to the fourth root of $\sigma\nu^2$.

THREE POSSIBLE NON-DIMENSIONAL COEFFICIENTS

It is possible to write down three non-dimensional coefficients:

$$C/c_{m1}; \quad C/u; \quad \text{and} \quad c_{m1}/u$$

If $C = \sqrt{2gH}$, $c_{m1} = \dfrac{Q}{\frac{\pi}{4}D^2}$, and $u = \dfrac{\pi nD}{60}$, these coefficients

[2] Marks' Mechanical Engineers' Handbook, p. 1151.

become

$$\left.\begin{aligned}\sigma &= \left(\frac{C}{c_{m1}}\right)^2 = \frac{2gH}{{c_{m1}}^2}\\ \nu &= \frac{Hg}{u^2}\end{aligned}\right\}\ \text{Coefficients of thrust}$$

$$\lambda = \frac{c_{m1}}{u}\quad \text{Coefficient of pitch}$$

Since $\lambda = \sqrt{2\frac{\nu}{\sigma}}$, only two of these three coefficients can be written down independently. If we select σ and ν, we get (since $c_{m1} = Q/(\pi D^2/4)$ and $u = \pi nD/60$):

$$\sigma = \frac{g\pi^2 HD^4}{8Q^2}$$

and

$$\nu = \frac{g}{\pi^2}\frac{60^2 H}{n^2 D}$$

A non-dimensional coefficient which shall fully determine the characteristics of a turbine should not contain the term D. Such a coefficient is $\sigma\nu^2$, since

$$\sigma\nu = \frac{\pi^2 g^3 60^4 H^3}{8Q^2 n^4 \pi^4} = \frac{240 \times 10^{11}}{n_s{}^4}$$

It may be convenient to use an expression

$$\psi = \sqrt[4]{\frac{1}{\sigma\nu^2}} = \frac{n_s}{2210}$$

The advantages of using $\sigma\nu^2$ are, first, that it is non-dimensional, and secondly, that it relates all types of hydrodynamic machinery. This should be particularly valuable in the case of the Kaplan propeller turbine.

Whether it is expedient to introduce this non-dimensional coefficient is ultimately a matter for practical engineers to decide.

Rack Structure and Headgates of Cedar Creek Hydroelectric Station

Details of the Design of the Rack Structure and of the Type of Head Gate Adopted for the Water Intake at This Station of the Duke Power Company on the Catawba River, Near Great Falls, S. C.

By W. S. LEE,[1] CHARLOTTE, N. C.

TO A CONSIDERABLE extent the successful operation of a water-power plant depends on the layout of the penstock intake, which for heads between 50 and 120 ft. is usually formed in the concrete masonry of the power-house bulkhead. To prevent floating debris from entering the intake the practice is to provide a steel rack in front of the entrance to the intake, and to be able to close the penstock inlet in case of emergency, or for making repairs to the turbine, a gate is installed.

In August, 1926, the Duke Power Company completed construction of its Cedar Creek Station on the Catawba River near Great Falls, S. C., having an installation of three 19,000-hp. turbines operating under a head of 60 ft., and this paper describes the design of the rack structure and the type of head gate adopted for the water intake.

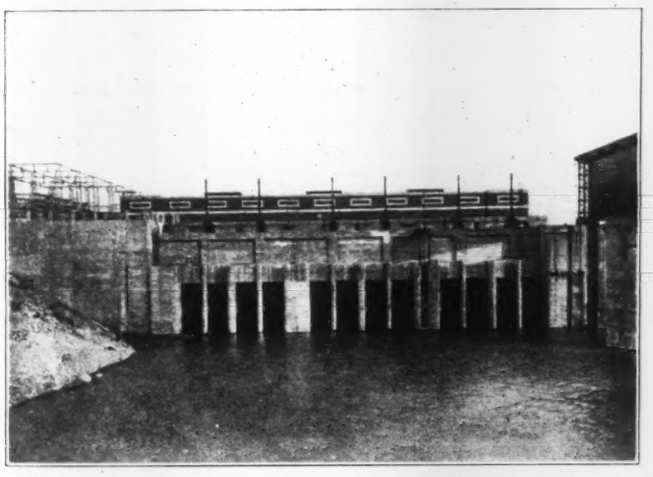

FIG. 1 VIEW OF POWER-HOUSE FOREBAY

THE RACK STRUCTURE

The rack structure forming the upstream portion of the power-house bulkhead contains twelve rack sections 9 ft. 3 in. wide by 39 ft. high, which are built of structural steel and arranged in a vertical position. As shown on the sectional plan of intake, Fig. 2, there are four sections in front of the water inlet to each turbine, and grooves are provided for the insertion of stop logs to take care of unwatering and permit of the inspection and occasional painting of the steelwork.

Considering the large capacity of the turbine, the spacing of the 6-in. wide by $^3/_8$-in-thick rack bars is made $4^1/_2$ in. on centers. The length and number of rack bars used were governed by following requirements:

1 The velocity of water through the rack openings should not be more than 2.5 ft. per sec. when the turbine is delivering 18,000 hp. at 0.9 gate opening and under a head of 60 ft., calling for a discharge of 2970 cu. ft. per sec. at 89 per cent guaranteed efficiency.

2 The maximum length of the rack bars should not exceed the rolled length of the merchant bars, and the cutting of them to the required length should be a minimum so as to eliminate waste of material.

The frame for each rack section is built up of heavy structural-steel members riveted together in the manufacturer's shop so as to facilitate their erection in the field. Slotted angle irons are riveted to the intermediate horizontal beams of each frame and a slotted spacing plate to the horizontal top beam of the frame. In the field the rack bars are then inserted into the slots of the spacing angles and plate and fastened at the top and bottom as shown in Fig. 3. A bent slotted spacing plate is riveted to the top connection for the purpose of providing a guide for the hand rake. It will be noticed that, with the exception of the cutting to the required length, the rack bars do not require any shop work. There are 24 rack bars in a frame, and each set of bars was shipped to the site in a bundle weighing 7200 lb.

The rack structure is designed to withstand hydrostatic pressure against its covered upstream face due to extreme high water and considering the penstock intake unwatered. In this case the steelwork will be stressed to its elastic limit. All the steel work is covered with two field coats of Valdura asphalt paint.

THE HEADGATES

Each penstock intake is divided into two openings by a 5-ft-thick partition of concrete masonry. These openings are controlled by rectangular gates of the butterfly type, 17 ft. 9 in. wide by 24 ft. 9 in. high, each having in the center a cast-steel girder 3 ft. in depth and 2 ft. 8 in. in width, with bronze-bushed trunnions 2 ft. 4 in. in outside diameter and 20 in. in length at each end. The wings of the gate are of cast iron and are attached to the center section by eighteen shrink links of $2^1/_2$-in. square section and T-headed at each end and twenty-four $2^1/_4$-in. machine bolts. A built-up cast-iron wall frame with cast-iron pillow blocks and cast-steel sole plates which rest on a grillage of I-beams embedded in the concrete masonry is provided for each gate.

The upper portion of the gate, which has no direct contact with the seat on the wall frame, has beveled surfaces, and bronze seals attached to the wall frame are provided to secure water-tightness of the gate.

The gate is designed to operate against a maximum head of water of 48 ft. above its center line of rotation acting on the upstream side with no back pressure on the downstream side. The distance between the centroid of water pressure and the center of rotation is slightly larger than is necessary to overcome the calculated friction in the trunnion bearings, and in addition 1-in. deformed steel bars are cast into the pocket of the lower wing which is to be filled with the necessary concrete after erection of the gate so as to cause the open gate to close if the operating mechanism is disconnected. The general design of the gate and wall frame, including its sealing devices, is shown in Fig. 5. The connecting rod, consisting of a 9-in. extra heavy lap-welded wrought-steel pipe, has a cast-steel pin-connecting piece attached to each end; the lower pin connection is bronze-bushed and fitted with a $4^1/_2$-in-diameter steel pin for attaching to the gate and the upper pin connection is slotted for and equipped with a $4^1/_2$-in-diameter steel pin with a 4-in. allowance for overtravel. This device forms part of an arrangement by which a buckling of the connecting rod is prevented when the gate is reaching its lowest position, as will be more fully described later. A cast-steel crosshead moving vertically in a cast-steel crosshead guide anchored to the concrete work of the intake forms the connection between the 9-in. steel pipe and the 5-in. outside-diameter steel screw stem of the operating hoist.

As shown in Fig. 6, the gate is operated by an individual, completely enclosed hoist located on top of the intake structure above the extreme highwater level. Referring to Fig. 7, the screw stem is

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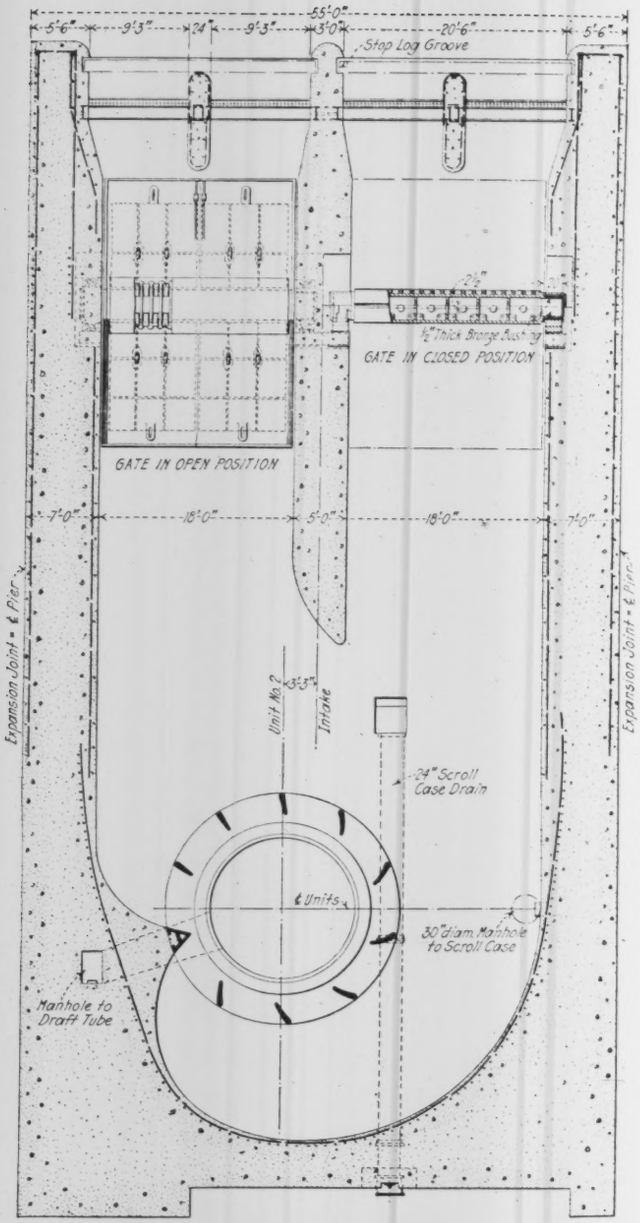

FIG. 2 SECTIONAL PLAN OF INTAKE

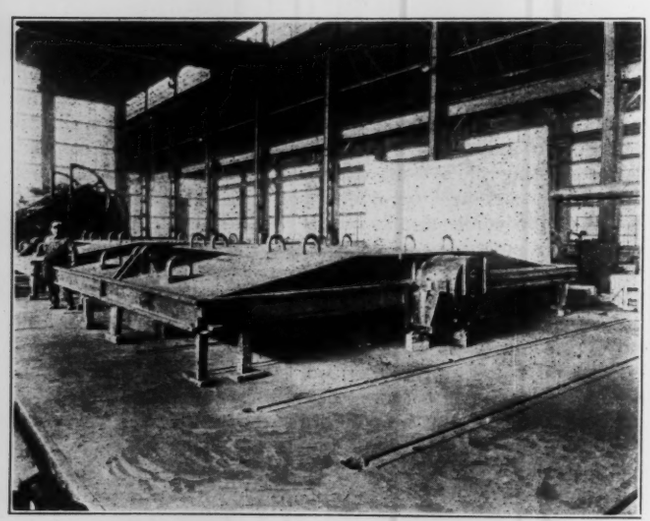

FIG. 4 VIEW OF ASSEMBLED GATE IN MANUFACTURER'S SHOP

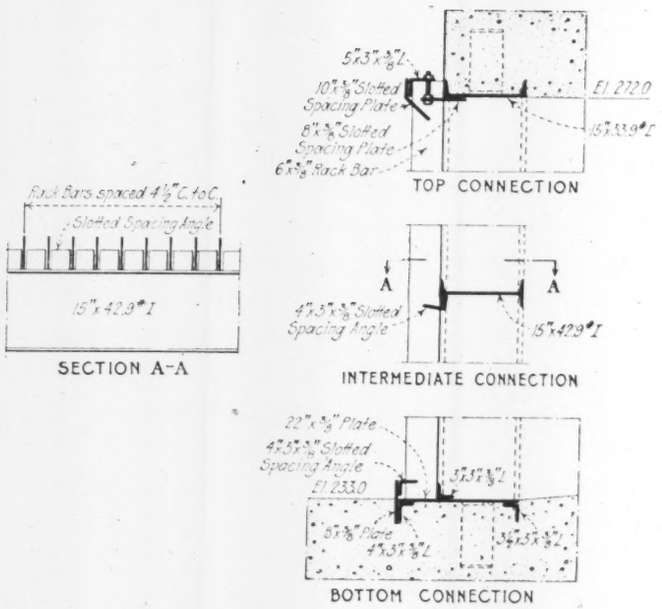

FIG. 3 DETAILS OF RACK AND FRAME CONNECTIONS

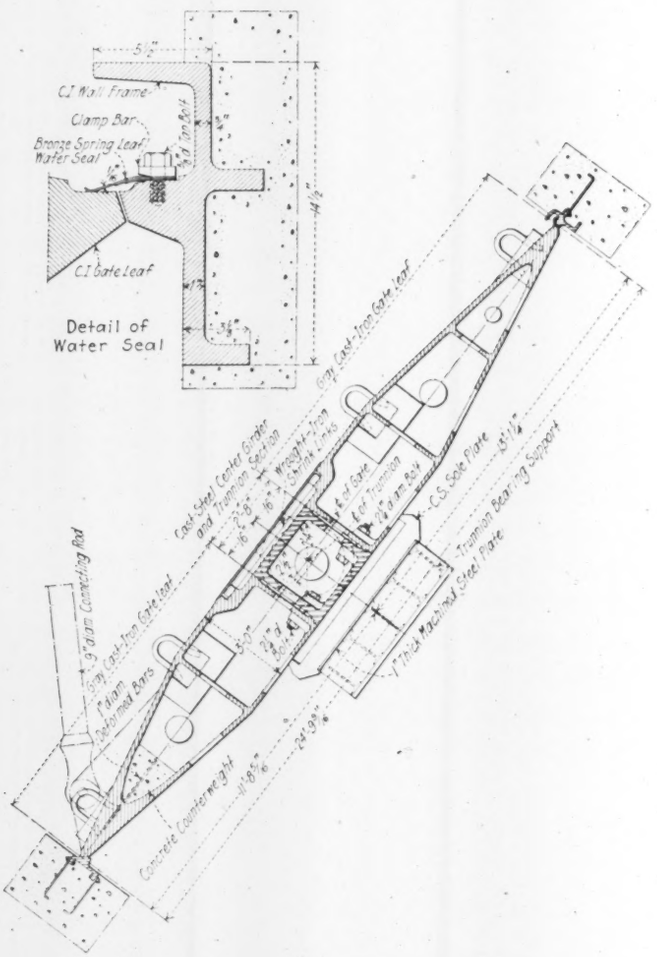

FIG. 5 LONGITUDINAL SECTION THROUGH GATE AND DETAIL OF WATER SEAL

raised or lowered by means of a manganese-bronze operating nut which is keyed to a cast-steel bevel-gear drive so arranged that the nut can slide upward in the bevel gear in case the limit switch controlling the downward stroke of the screw stem fails to operate. A nest of heavy steel coil springs will then resist the upward travel of the nut and gradually create the necessary overload, causing

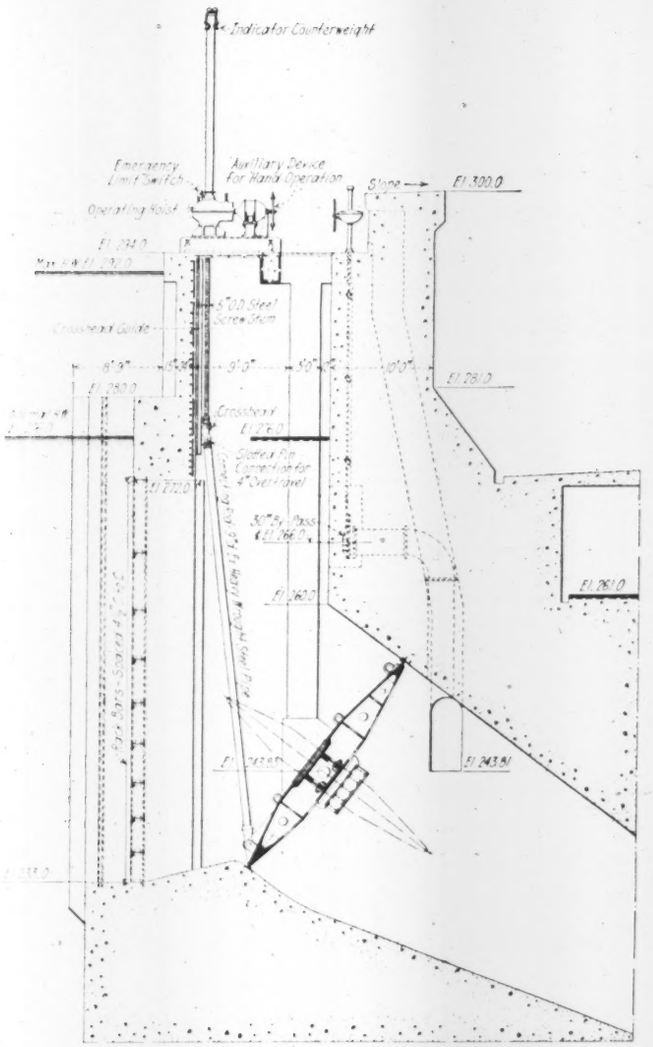

FIG. 6 SECTION THROUGH INTAKE SHOWING HEADGATE AND OPERATING DEVICE

a tripping of the overload release switch of the motor. An allowance of 15 in. for overtravel on the upward stroke is made by proportioning the length of the screw stem and crosshead guide. The 9-in. steel-pipe cover for the screw stem attached to the top of the gear housing is provided with a gate-position indicator.

Direct-current, 230-volt, electric power is supplied for the operation of the hoists. Each headgate motor is completely enclosed and has a starting torque of 500 ft-lb., decreasing immediately to 380 ft-lb. during 600 revolutions, the running torque being 125 ft-lb. The operating mechanism is designed to open or close the gate in approximately eight minutes, and it is capable of developing an upward lift of 150,000 lb. and a downward push of 50,000 lb. in the vertical screw stem. A weatherproof push-button control station is provided for each hoisting mechanism on top of the bulkhead, and a non-weatherproof control station on the power-house benchboard for distant closing of each gate.

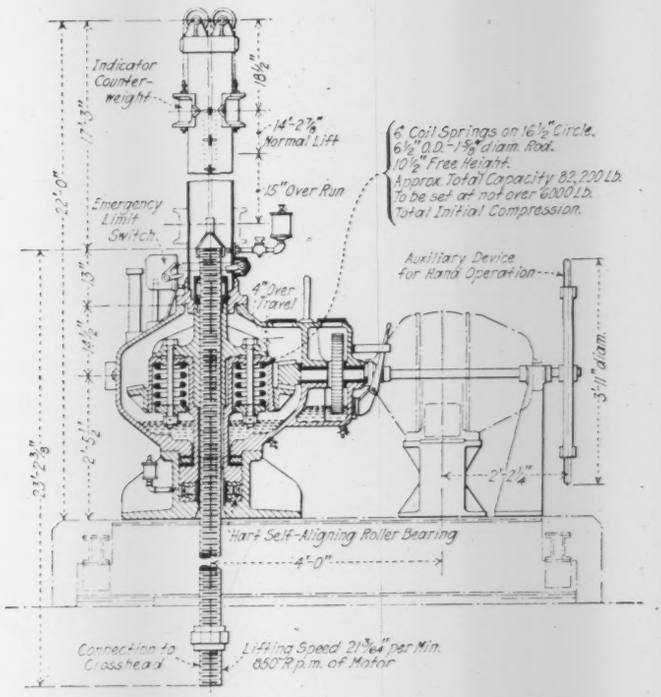

FIG. 7 SECTIONAL ELEVATION OF OPERATING HOIST

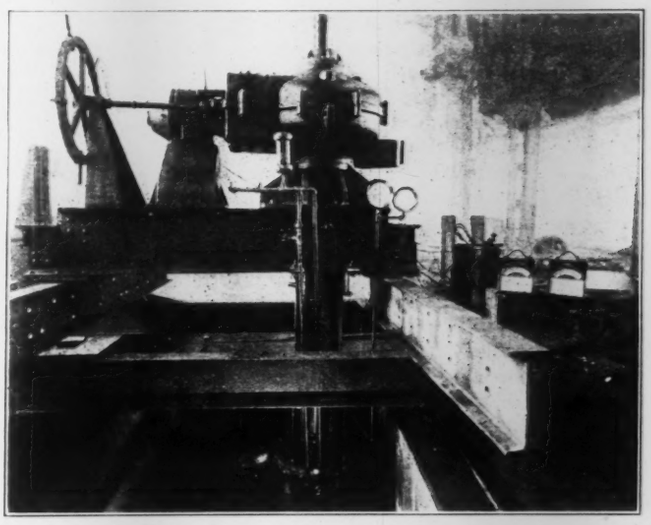

FIG. 8 VIEW OF GATE HOIST ON TESTING BLOCK IN MANUFACTURER'S SHOP

The emergency push-button control on the benchboard is provided to take care of a possible runaway of the turbine, as the operating hoist is designed to close the gate under the condition of full discharge of water flowing through the passageways of the turbine; normally the gates are operated from the control station on top of the bulkhead.

A 30-in-diameter bypass is provided in each intake for filling the flume before the headgates are opened and the turbine started.

Comparison and Limitations of Various Water-Hammer Theories

Confirmation of Accuracy of Elastic-Water-Column Theory by Experimental Test Data—Charts for Graphically Solving Problems of Maximum Pressure Rise with Uniform Gate Motion and Complete Closure

By RAY S. QUICK,[1] SAN FRANCISCO, CAL.

THIS paper makes a general comparison between the principal theories and formulas for water hammer produced near the outlet of a closed conduit by closure of the discharge orifice or gate.

The accuracy of the elastic-water-column theory is emphasized and confirmed by experimental test data, and charts are given for graphically solving problems of maximum pressure rise with uniform gate motion and complete closure.

Water-hammer theories advocated during recent years may be divided into two classes: those which consider the elastic properties of the water and conduit walls, and those which neglect elasticity and instead treat the water column and pipe as rigid and apply the laws of motion direct.

The comparison of these theories will be made in the following general terms.

Nomenclature

a = velocity of pressure wave along pipe, ft. per sec.
A = cross-sectional area of pipe, sq. ft.
b = thickness of pipe walls, in.
d = inside diameter of pipe, in.
E = Young's modulus for pipe walls, lb. per sq. in.
g = acceleration of gravity, ft. per sec. per sec.
h = pressure rise or excess head above normal, ft.
h_{max} = pressure rise of instantaneous closure = aV_0/g, ft.
H_0 = initial steady head near gate, corresponding to V_0, ft.
k = bulk modulus of elasticity of water, lb. per sq. in.
K = pipe-line constant = $\frac{h_{max}}{2H_0} = \frac{aV_0}{2gH_0}$ = Rho (P) of Allievi
l = length of any uniform section of pipe, ft.
L = length of pipe from gate to forebay or other point of relief, ft.
N = time constant or number of $2L/a$ intervals in time of closure = $aT/2L$ = Theta (Θ) of Allievi
Sub (n) = designation of the interval, measured in steps of $2L/a$ seconds from start of gate closure when applied to terms h, P, V, and ϕ. Refers also to physical pipe-line section characteristics A, a, and l.
P = pressure rise as a proportion of $h_{max} = \frac{h}{\frac{aV_0}{g}} = \frac{gh}{aV_0}$
Q_0 = initial steady flow in pipe prior to start of gate closure corresponding to H_0, cu. ft. per sec.
T = time of gate-closure travel, sec.
V_0 = velocity in pipe near gate, corresponding to H_0 and Q_0, ft. per sec.
V_n = velocity in pipe near gate at any time designated as n, ft. per sec.
Σ = a summation of consecutive like terms, in steps of $2L/a$ each, back from point under consideration to start of closure. Refers particularly to $\Sigma(P)$, as in Equation [7]
ϕ_0 = initial gate-opening factor, where $V_0 = \phi_0 \sqrt{H_0}$
ϕ_n = proportion of initial gate factor ϕ_0, at any time designated as sub (n).

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The Elastic-Water-Column Theory

The elastic-water-column theory, which considers the elasticity of both the water and pipe, and which most accurately interprets experimental data, has been investigated and reported in detail by Joukovsky,[2] Allievi,[3] N. R. Gibson,[4] R. D. Johnson,[5] and W. F. Durand.[6] For the present comparison, an ideal pipe line of the following characteristics will be assumed, the limitations imposed being discussed later.

a Without friction to flow.
b Of uniform diameter, thickness, and material.
c Without branches, standpipes, or other special features.
d Travel of pressure waves without loss in velocity or amplitude and with perfect reflection at gate and forebay.

Joukovsky found that the pressure rise on instantaneous closure was proportional to the initial velocity destroyed and to the elasticity of the water and pipe walls, and was expressed by

$$h_{max} = \frac{aV_0}{g} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

This pressure rise traveled to the forebay and back, in time $2L/a$, and then was converted into a subnormal wave of the same magnitude which in turn made a round trip to the forebay before reconversion into a supernormal pressure wave identical with the initial one.

Gradual gate closure may be considered the equivalent of a series of consecutive instantaneous closure movements, each producing a pressure wave proportional to the velocity destroyed, and traveling between gate and forebay; the summation representing the net pressure change at any time. Thus the pressure will continue to rise at the gate in direct proportion to the velocity destroyed until reflection occurs after $2L/a$ seconds to modify the total, in the manner expressed by

$$h_n - h_{n-1} = \frac{a}{g}(V_{n-1} - V_n) - 2h_{n-1} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

which designates the rise in pressure during any interval of time of $2L/a$ seconds.

The summation of these increments expressed by Equation [2], from the start of gate closure to the end of the nth interval, may be secured by adding, as follows:

$$h_1 - h_0 = \frac{a}{g}(V_0 - V_1) - 2h_0 \quad \ldots\ldots\ldots\ldots \text{for first interval}$$

$$h_2 - h_1 = \frac{a}{g}(V_1 - V_2) - 2h_1 \quad \ldots\ldots\ldots\ldots \text{for second interval}$$

$$h_3 - h_2 = \frac{a}{g}(V_2 - V_3) - 2h_2 \quad \ldots\ldots\ldots\ldots \text{for third interval}$$

$$\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots$$

$$h_n - h_{n-1} = \frac{a}{g}(V_{n-1} - V_n) - 2h_{n-1} \quad \ldots\ldots\ldots\ldots \text{for } n\text{th interval}$$

adding,

$$h_n - h_0 = \frac{a}{g}(V_0 - V_n) - 2(h_0 + h_1 + h_2 + \ldots + h_{n-1}), \text{ or}$$

[2] Proc. Am. W. W. Assn., 1904, p. 341.
[3] Theory of Water Hammer. Translated by E. E. Halmos, 1925.
[4] Trans. A.S.C.E., vol. 83 (1920), p. 707.
[5] Trans. A.S.C.E., vol. 79 (1915), p. 277; also vol. 83 (1920), p. 754.
[6] Hydraulics of Pipelines, 1921.

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$$h_n - h_0 = h_{max} - \frac{a}{g} V_n - 2\Sigma(h_{n-1}) \quad [3]$$

As the pressure rise starts with gate travel, $h_0 = 0$. When full closure is reached, $V_n = 0$, so the final pressure rise at the instant of complete closure is

$$h_n = h_{max} - 2\Sigma (h_{n-1}) \quad [4]$$

In other words, the final rise equals the rise of instantaneous closure less twice the summation of the rises existing at all previous intervals measured consecutively backward in steps of $2L/a$ each. Closure in one interval or less reduces $\Sigma(h_{n-1})$ to zero, making $h_n = h_{max}$.

The pressure rise h_n may be determined for any proportion of the nth interval by determining $\Sigma(h_{n-1})$, by adding the preceding rises measured for the same proportion of each interval or, in other words, measured back in steps of $2L/a$ each from the point under consideration.

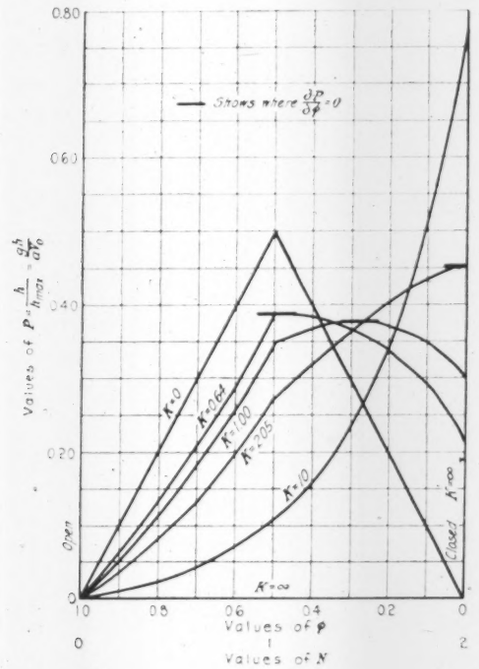

FIG. 1 PRESSURE-RISE-TIME CHART SHOWING EFFECT OF VARIATION IN VALUE OF K
(Uniform gate motion and complete closure in two intervals.)

Equation [3] contains two variables, h and V, and therefore requires a second and independent relation for a solution; this being determined from the law governing the discharge from an orifice, expressed as

$$V_n = V_0 \phi_n \sqrt{\frac{H_0 + h_n}{H_0}} \quad [5]$$

By combining Equations [3] and [5], and eliminating V_n, there results

$$h_n = h_{max} - 2\Sigma (h_{n-1}) - h_{max}\phi_n \sqrt{\frac{H_0 + h_n}{H_0}}$$

Dividing through by h_{max} and writing $\frac{h_n}{h_{max}}$ as P_n,

$$P_n = 1 - 2\Sigma - \phi_n \sqrt{1 + 2KP_n} \quad [6]$$

Collecting, squaring, and solving for P_n,

$$P_n = 1 - 2\Sigma + \phi^2_n K \pm \phi_n \sqrt{1 + 2K(1 - 2\Sigma) + \phi^2_n K^2} \quad [7]$$

The *minus* sign only should be used before the radical as the plus sign may make P_n greater than unity, which of course is contrary to fact and to Equation [3].

Equation [7] is the general formula for calculating the pressure-rise-time curve for any gate motion and initial conditions, based on the elastic-column theory. Uniform gate motion is an average general case and permits expressing ϕ in terms of N, making a chart solution simple. Any non-uniform gate motion may be determined to suit a special case, and Equation [7] applied direct. The characteristics of the pressure-rise-time curves as a solution of Equation [7], for the special case of uniform gate motion and complete closure, are indicated by Figs. 1 and 2. The first considers closure in two intervals, with various values of K from zero to infinity, while the second considers K as always unity and closure from instantaneous to an infinite number of intervals. Each figure represents a family of curves and shows the influence of initial conditions upon the form of the curve.

The maximum rise in pressure is often of more importance to the engineer than the form of the curve producing it. For closure in two intervals as in Fig. 1, this maximum occurs at the end of the first interval for all values of K from 0 to 0.64, during the second interval for all values from 0.64 to 2.05, and on complete closure for all greater values, reaching a maximum value of unity for P when K is infinite. These maximum values may be confirmed by differential calculus by finding values of ϕ and K which satisfy the relation $\frac{\partial P}{\partial \phi} = 0$. The practical application of this method to the third and later intervals is quite tedious as the term Σ enters. Fig. 3 shows only the maximum values of P, for uniform gate motion and complete closure, and therefore becomes a general chart for

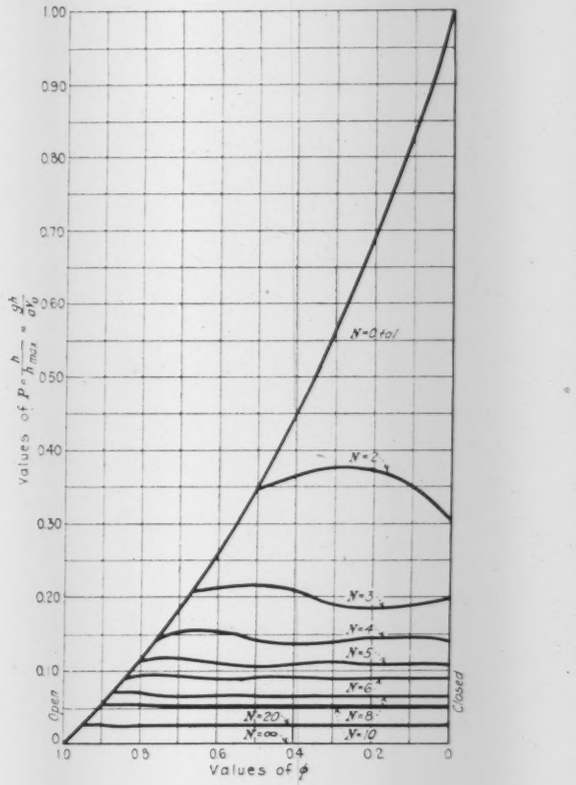

FIG. 2 PRESSURE-RISE-TIME CHART SHOWING EFFECT OF VARIATION IN VALUE OF N
(Complete closure with uniform gate motion. $K = 1$.)

the graphical solution of such problems. It covers closure from instantaneous to fifty intervals, and values of K from 0.07 to 40, which should be ample for the great majority of practical cases. If the chart be extended to the left to $K = 0$, P approaches $1/N$ as a limit. If it be extended to the right indefinitely, P approaches unity as a limit. Extended downward, P finally becomes zero. A study of Fig. 7, mentioned later, is of great interest in this subject of limiting values as N is increased. It is apparent that a high-head plant with K less than unity will experience maximum water hammer at the end of the first interval, while a low-head installation with K greater than unity may not reach the full pressure rise until some later interval. Therefore retardation of the rate of closure of the gate, near final closure, may reduce the maximum rise in a low-head plant and be of little or no assistance in a high-

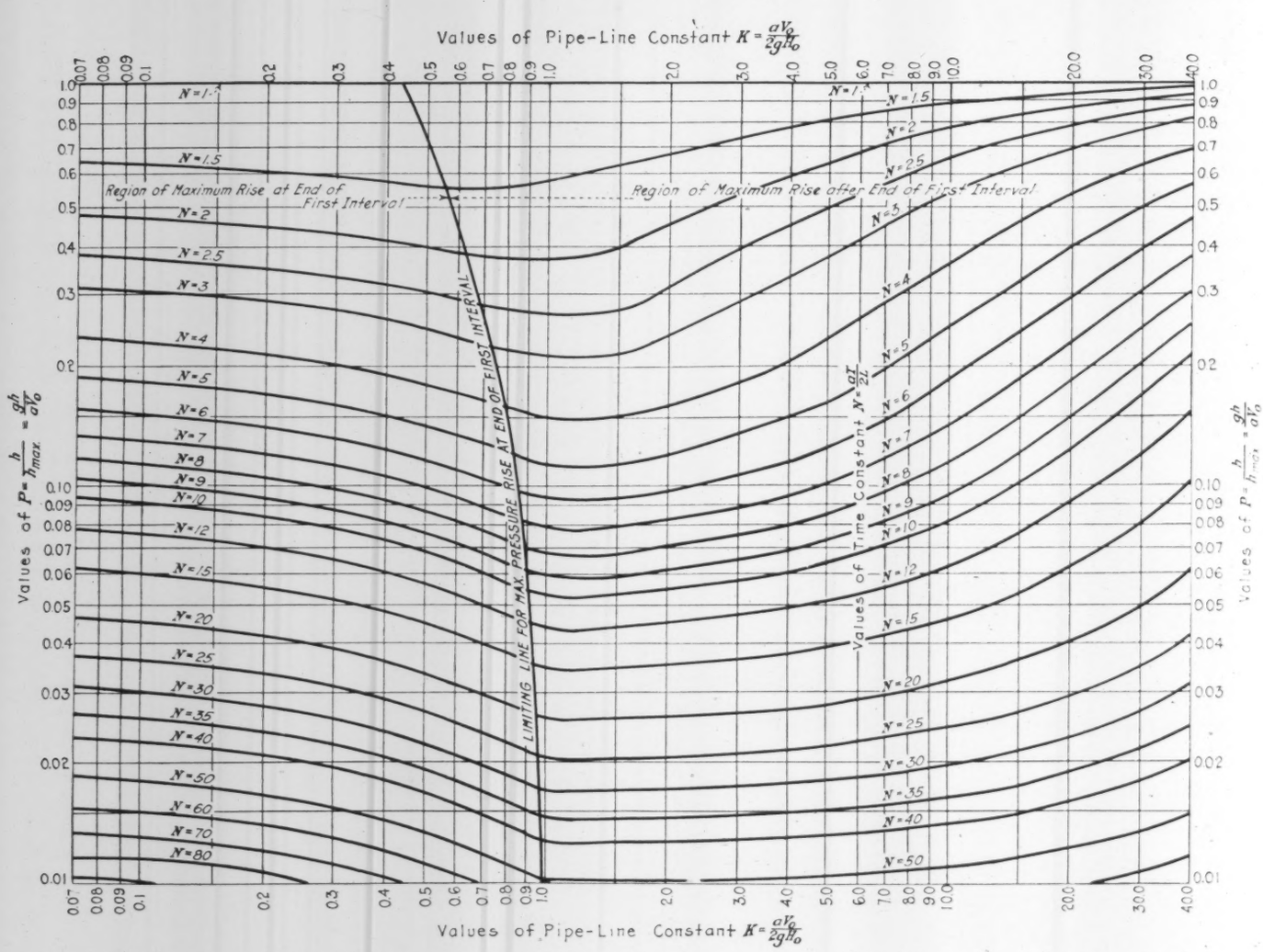

FIG. 3 CHART SHOWING MAXIMUM PRESSURE RISE WITH UNIFORM GATE MOTION AND COMPLETE CLOSURE—BASED ON ELASTIC-WATER-COLUMN THEORY

(NOTE: Ratio of pressure rise h to initial steady head H_0 determined from relation $2KP = h/H_0$.)

head installation, depending upon the number of intervals for gate travel. With closure in a constant number of intervals, the pressure rise becomes greater in proportion to the initial head as K increases, indicating that water hammer becomes more important relatively in low-head installations. The rate of travel of the pressure rise a may be determined for steel pipe from Fig. 4.

Allievi's[7] chart for maximum pressure rise with uniform gate motion and complete closure, based upon the elastic-water-column theory, appears in slightly modified form in Fig. 5. The solid lines indicate the ratio of the total gate pressure to the initial head, while the dotted lines show the interval as well, where the maximum pressure is first reached; this being of importance in determining the approximate form of the pressure-rise–time curve and the proportion of the maximum reached in the case of partial gate closure. The value of a for use with Fig. 5 may be determined for steel pipe from Fig. 4.

APPROXIMATE FORMULAS

As the more generally known approximate formulas fail to check each other and the elastic-column formula as well, except in isolated cases, it is of interest to compare them in general common terms to fix their limits of safe use.

H. C. Vensano,[8] basing his conclusions on the work of Joukovsky,[9] conformed to the elastic-column theory with the exception that no allowance was made for the change in discharge and velocity due to the variation in total head at the gate. In other words, Equation [5] would be modified to read $V_n = V_0\phi_n$. This coincides

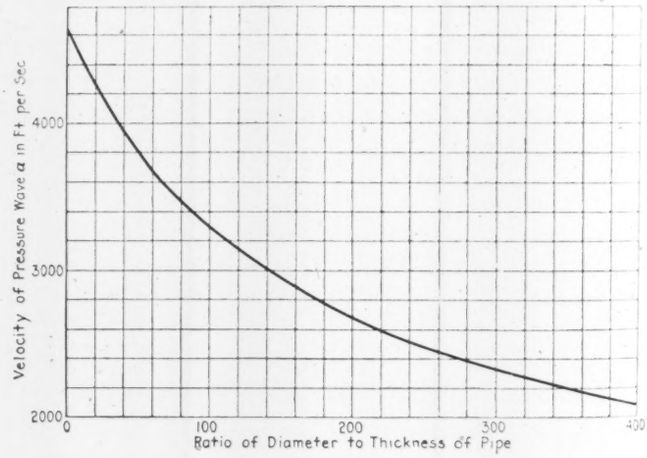

FIG. 4 CHART SHOWING VELOCITY OF TRAVEL OF PRESSURE WAVE IN ELASTIC WATER COLUMN

Formula: $a = \frac{4660}{\sqrt{1 + \frac{kd}{Eb}}} = \frac{4660}{\sqrt{1 + \frac{d}{100b}}}$

where a = velocity of travel of pressure wave, ft. per sec.
k = bulk modulus of elasticity of water = 294,000 lb. per sq. in.
E = Young's modulus for pipe walls = 29,400,000 lb. per sq. in. approx. for steel
d = inside diameter of pipe, in.
b = thickness of pipe walls, in.

[7] Theory of Water Hammer, L. Allievi. Translated by E. E. Halmos, 925.

[8] Trans. A.S.C.E., vol. 79 (1915), p. 289.

[9] Proc. Am. W. W. Assn., 1904, p. 341.

fully with the elastic-column theory only when $\sqrt{\frac{H_0 + h_n}{H_0}} = 1$ or when $K = 0$, and is approached as a limit in very high-head installations. With uniform gate motion and complete closure, the maximum rise in pressure is given by Vensano as

$$\left.\begin{aligned} h &= \frac{2LV}{gT}\left(\text{when } T \geqq \frac{2L}{a}\right) \\ &= \frac{aV_0}{g}\left(\text{when } T < \frac{2L}{a}\right) \end{aligned}\right\} \quad \ldots [8]$$

Conversion into terms of P and N reduces [8] to

$$P = \frac{1}{N} \quad \ldots [9]$$

M. M. Warren,[10] on the assumptions of uniform pressure rise during the first interval of closure and absence of subnormal reflected waves at the gate, developed the following expression for maximum pressure rise.

$$\left.\begin{aligned} h &= \frac{LV}{g\left(T - \frac{L}{a}\right)}\left(\text{when } T \geqq \frac{2L}{a}\right) \\ &= \frac{aV_0}{g}\left(\text{when } T < \frac{2L}{a}\right) \end{aligned}\right\} \quad \ldots [10]$$

Expressed in terms of P and N, this reduces to

$$P = \frac{1}{2N - 1} \quad \ldots [11]$$

When N is large compared with unity, this further reduces to the limit of $P = 1/2N$, which is half the value given by Vensanos' Equation [9].

The rigid-water-column theory neglects elasticity but considers the influence of the pressure rise upon the total head and discharge at the gate. It has been investigated by Allievi,[11] R. D. Johnson,[12] A. H. Gibson,[13] and others, and is found in many different forms. The work of R. D. Johnson[12] is particularly interesting as it develops not only the formulas for the form of the pressure-rise-time curve and its maximum, but also the limits of accurate application in comparison with the elastic-column theory; setting forth the principal information in the form of charts. This theory may be expressed by two simultaneous equations, the first being derived from the force equation of motion, and the second from the discharge from an orifice as by Equation [5], as follows:

$$h = -\frac{L}{g}\frac{dv}{dt} \quad \ldots [12]$$

[10] Trans. A.S.C.E., vol. 79 (1915), p. 238.
[11] Theory of Water Hammer. Translated by E. E. Halmos, 1925.
[12] Trans. A.S.C.E., vol. 79 (1915), p. 277; also vol. 83 (1920), p. 754.
[13] Hydraulics and Its Applications, 1919.

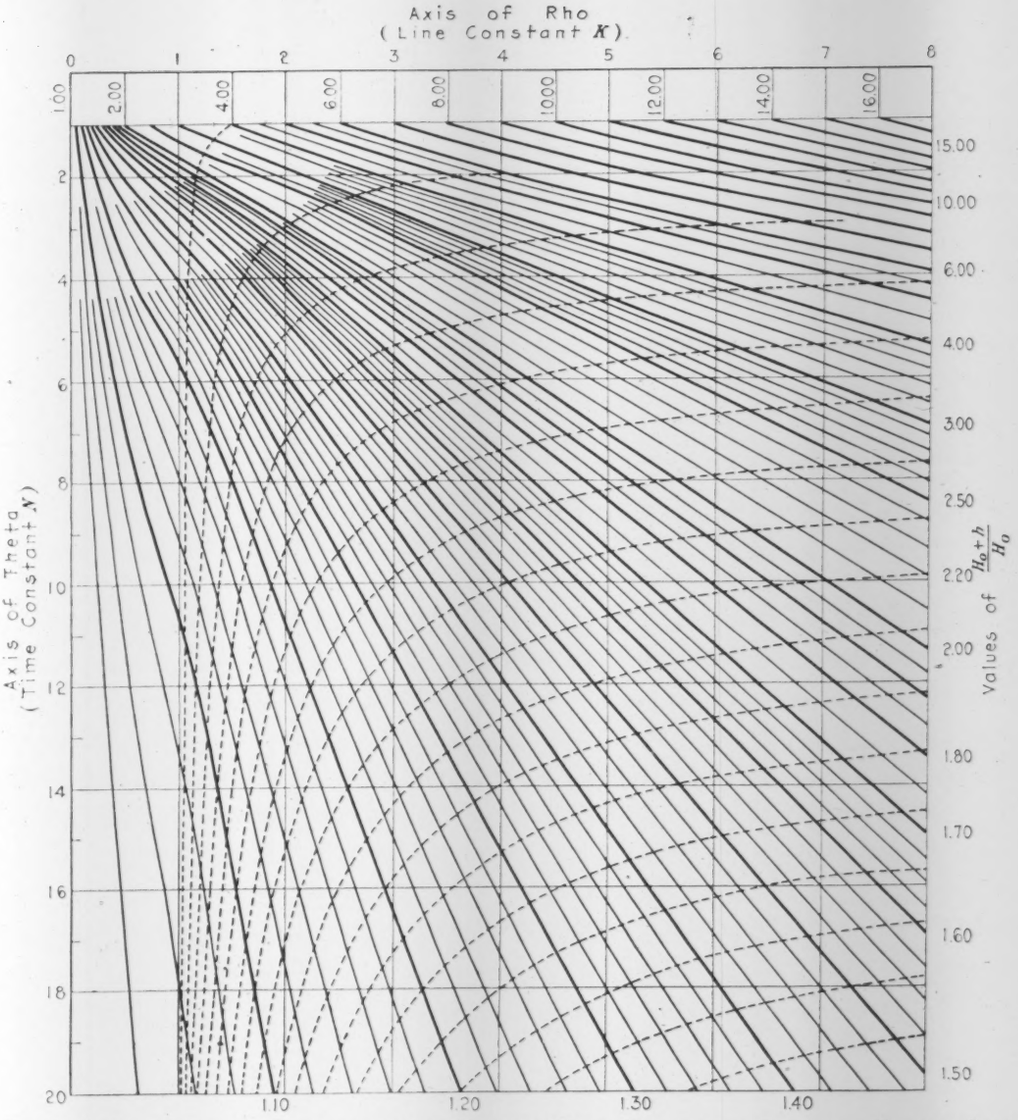

FIG. 5 ALLIEVI CHART FOR DETERMINATION OF MAXIMUM TOTAL PRESSURE AT GATE WITH UNIFORM GATE MOTION AND COMPLETE CLOSURE

[Solid diagonal lines give ratio $(H_0 + h)/H_0$; dotted lines (numbered S_1 to S_{17} from left) give interval in which maximum pressure is first reached.]

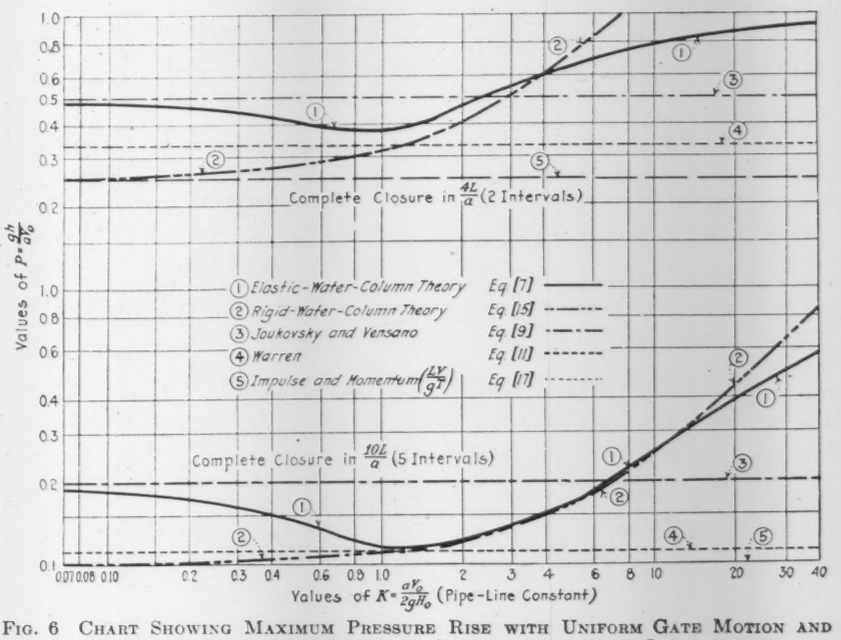
Complete Closure in $\frac{4L}{a}$ (2 Intervals)
Complete Closure in $\frac{10L}{a}$ (5 Intervals)
① Elastic-Water-Column Theory Eq [7]
② Rigid-Water-Column Theory Eq [5]
③ Joukovsky and Vensano Eq [9]
④ Warren Eq [11]
⑤ Impulse and Momentum $\left(\frac{LV}{g^2}\right)$ Eq [17]
Values of $P = \frac{gh}{aV_0}$
Values of $K = \frac{aV_0}{2gH_0}$ (Pipe-Line Constant)

FIG. 6 CHART SHOWING MAXIMUM PRESSURE RISE WITH UNIFORM GATE MOTION AND VARIOUS THEORIES FOR COMPLETE CLOSURE IN 2 AND 5 INTERVALS

$$V_n = V_0\phi_n\sqrt{\frac{H_0 + h_n}{H_0}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [13]$$

For the case of uniform gate motion, ϕ_n may be expressed in terms of time, the solution resulting in a logarithmic curve; while the maximum, occurring on complete closure, is given by

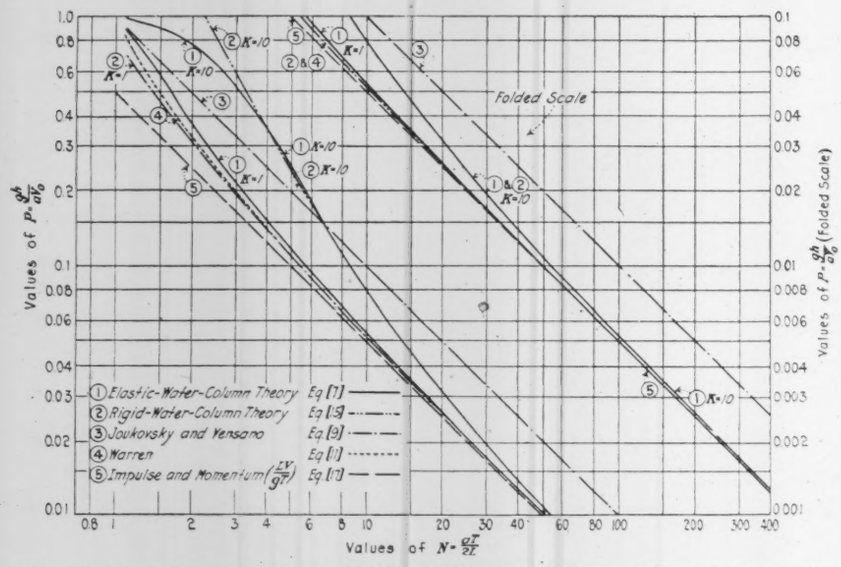
Folded Scale
Values of $P = \frac{gh}{aV_0}$
Values of $P = \frac{gh}{aV_0}$ (Folded Scale)
Values of $N = \frac{gT}{2L}$
① Elastic-Water-Column Theory Eq [1]
② Rigid-Water-Column Theory Eq [5]
③ Joukovsky and Vensano Eq [9]
④ Warren Eq [11]
⑤ Impulse and Momentum $(\frac{LV}{gT})$ Eq [17]

FIG. 7 CHART SHOWING MAXIMUM PRESSURE RISE WITH UNIFORM GATE CLOSURE AND VARIOUS THEORIES

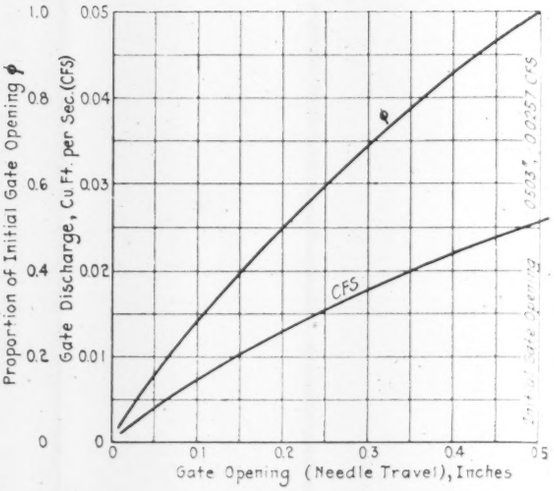

FIG. 10 OBSERVED TEST CHARACTERISTICS OF BIG CREEK EXPERIMENTAL PENSTOCK, SOUTHERN CALIFORNIA EDISON CO., MAY, 1926

(Static head, 306.7 ft.; net head with 0.0257 cu. ft. per sec. flow, 301.6 ft.; length of pipe line, 3060 ft.; inside diameter, 2.06 in.; velocity with 0.0257 cu. ft. per sec. flow, 1.11 ft. per sec.; period $2L/a$ = 1.40 sec.; a = 4371 ft. per sec.; K (line constant) = $aV_0/2gH_0$ = 0.250, $h_{max} = aV_0/g$ = 151 ft. Needle travels at uniform rate.)

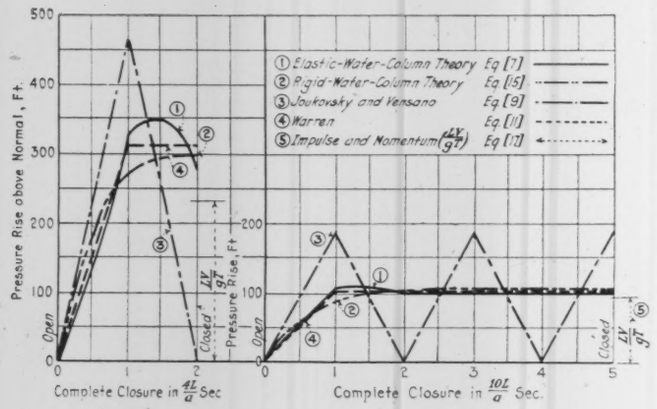
Pressure Rise above Normal, Ft.
Pressure Rise, Ft.
① Elastic-Water-Column Theory Eq [1]
② Rigid-Water-Column Theory Eq [5]
③ Joukovsky and Vensano Eq [9]
④ Warren Eq [11]
⑤ Impulse and Momentum $(\frac{LV}{gT})$ Eq [17]
Open
Closed
Complete Closure in $\frac{4L}{a}$ Sec.
Complete Closure in $\frac{10L}{a}$ Sec.

FIG. 8 PRESSURE-RISE–TIME CHART SHOWING EFFECT OF RATE OF CLOSURE UPON PRESSURE RISE WITH UNIFORM GATE MOTION AND VARIOUS THEORIES

(Conditions: a = 3000 ft. per sec.; V_0 = 10 ft. per sec.; L = 1500 ft.; H_0 = 465 ft.; $2L/a$ = 1 sec.; $K = aV_0/2gH_0$ = 1; $N = aT/2L$ = 2 and 5.)

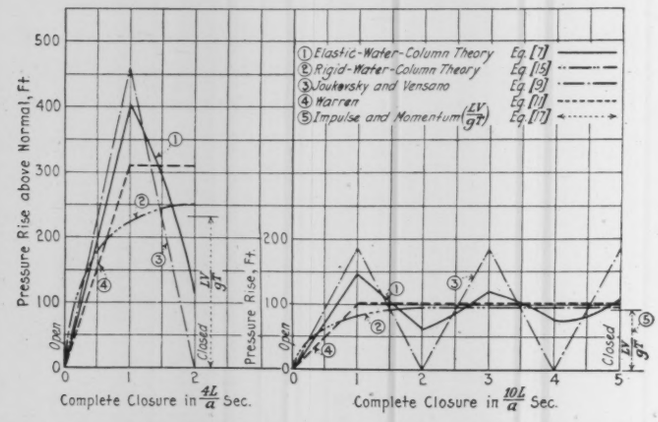
Pressure Rise above Normal, Ft.
Pressure Rise, Ft.
① Elastic-Water-Column Theory Eq [1]
② Rigid-Water-Column Theory Eq [5]
③ Joukovsky and Vensano Eq [9]
④ Warren Eq [11]
⑤ Impulse and Momentum $(\frac{LV}{gT})$ Eq [17]
Open
Closed
Complete Closure in $\frac{4L}{a}$ Sec.
Complete Closure in $\frac{10L}{a}$ Sec.

FIG. 9 PRESSURE-RISE–TIME CHART SHOWING EFFECT OF RATE OF CLOSURE UPON PRESSURE RISE WITH UNIFORM GATE MOTION AND VARIOUS THEORIES

(Conditions: a = 3000 ft. per sec.; V_0 = 10 ft. per sec.; L = 4500 ft.; H_0 = 1500 ft.; $2L/a$ = 3 sec.; $K = aV_0/2gH_0$ = 0.31; $N = aT/2L$ = 2 and 5.)

$$h = \frac{LV_0}{2g^2H_0T^2}\left[LV_0 + \sqrt{4g^2H_0^2T^2 + L^2V_0^2}\right] \quad \ldots\ldots [14]$$

which, on conversion into terms of P, K, and N, becomes

$$P = \frac{K + \sqrt{K^2 + 4N^2}}{4N^2} \quad \ldots\ldots\ldots\ldots [15]$$

Under high-head conditions where K is very small in value, or where closure is slow, thus making N large in comparison with K, P approaches a limiting value of $1/2N$. It is very interesting to note at this point that Equation [7] can be reduced to Equation [15] in the special case where $P_{n-1} = P_n$. This occurs only for values of K greater than unity, the region increasing with N.

The simplest expression for water hammer, widely used for approximate work, is derived from the relation between impulse and change in momentum and is expressed by

$$h = \frac{LV}{gt} \quad \ldots\ldots\ldots\ldots\ldots\ldots [16]$$

Converted into terms of P and N it becomes

$$P = \frac{1}{2N} \quad \ldots\ldots\ldots\ldots\ldots\ldots [17]$$

which is identical with limiting values of Equations [11] and [15].

Graphical comparison is made in Figs. 6 and 7 between the preceding formulas for maximum pressure rise with uniform gate motion and complete closure. The rigid-column theory shows a close comparison for certain limiting values of K greater than unity, while the other formulas check only in a limited number of isolated cases. For small values of K compared with unity, the actual pressure rise may be twice that of certain of the approximate formulas, while for large values of K conditions may be reversed and the ratio much greater. As N is increased, and for K near unity, all of the curves converge to a common value of $1/2N$, with the exception of Vensano's Equation [9], which remains at $1/N$ or twice as great.

The forms of corresponding pressure-rise–time curves are given by Figs. 8 and 9 for two values of K, indicating the variation which may be expected in these specific cases.

EXPERIMENTAL DATA

A direct comparison between accurate experimental pressure-

rise-time curves for sudden and gradual gate closure, and the elastic-column theory of Equation [7], is shown in Figs. 11, 12, and 13, the gate calibration being shown by Fig. 10. The agreement in all cases is so close as to establish the accuracy of Equation [7] for the conditions of this particular pipe line, and to exclude the approximate formulas.

The calculations of the theoretical pressure-rise-time curves for Figs. 11, 12, and 13 were made from Equation [7] as follows:

FIG. 11: This is a condition of practically instantaneous closure, thus establishing the maximum at h_{max} for the full interval less the time required for gate closure. Therefore

$$h = h_{max} = \frac{aV_0}{g} = 4871\,\frac{0.55}{32.2} = 74.6 \text{ ft.}$$

FIG. 12: This is closure in less than one interval, thus reaching h_{max} on complete closure and maintaining it until reflected waves appear at the end of the interval.

During closure, Equation [7] reads:

$$P_1 = 1 + \phi_1^2 K - \phi_1 \sqrt{1 + 2K + \phi_1^2 K^2}$$

With $K = 0.25$, ϕ from Fig. 10, $T = 1.00$ sec., and $2L/a = 1.40$ sec.,

t (sec.)	0.25	0.50	0.75	1.00	1.40
ϕ_1	0.82	0.60	0.34	0.00	0.00
$1+\phi_1^2K$	1.168	1.090	1.029	1.000	1.000
$1+2K+\phi_1^2K^2$	1.542	1.522	1.507	1.500	1.500
$\phi\sqrt{1+2K+\phi_1^2K^2}$	1.018	0.741	0.417	0.000	0.000
P_1	0.150	0.349	0.612	1.000	1.000
h (ft.) = 151 P_1	22.7	52.7	92.4	151	151

After closure of gate, $P_n = 1 - 2\Sigma$ (as $\phi_n = 0$), and

t (sec.)	1.65	1.90	2.15	2.40	2.80
2Σ	0.300	0.698	1.224	2.000	2.000
$P_n = 1-2\Sigma$	0.700	0.302	−0.224	−1.000	−1.000
h_n (ft.) = 151 P_n	106	45.6	−33.8	−151	−151

The wave is symmetrical for subsequent intervals, the ordinates being the same in steps of $2L/a$ but alternately of opposite sign.

FIG. 13: The time of closure in this case is over two intervals, so h_{max} is not reached. During closure,

$$P_n = 1 - 2\Sigma + \phi_n^2 K - \phi_n \sqrt{1 + 2K(1-2\Sigma) + \phi_n^2 K^2} \text{ (Eq. [7])}$$

With $T = 3.00$ sec., $2L/a = 1.40$ sec., $K = 0.25$, and ϕ from Fig. 10,

n (interval)	←first→				←second→				←third
t (sec.)	0.20	0.50	0.90	1.40	1.60	1.90	2.30	2.80	3.00
ϕ (Fig. 10)	0.96	0.89	0.78	0.63	0.565	0.465	0.32	0.11	0.00
2Σ	0.000	0.000	0.000	0.000	0.066	0.180	0.370	0.642	0.716
$1-2\Sigma$	1.000	1.000	1.000	1.000	0.934	0.820	0.630	0.358	0.284
ϕ_n^2K	0.230	0.198	0.152	0.099	0.080	0.054	0.026	0.003	....
$1-2\Sigma+\phi_n^2K$	1.230	1.198	1.152	1.099	1.014	0.874	0.656	0.361	....
$2K(1-2\Sigma)$	0.500	0.500	0.500	0.500	0.467	0.410	0.315	0.179	....
$1+\phi_n^2K^2$	1.058	1.050	1.038	1.025	1.020	1.014	1.006	1.001	....
$1+2K(1-2\Sigma)+\phi_n^2K^2$	1.558	1.550	1.538	1.525	1.487	1.424	1.321	1.180	....
$\phi\sqrt{1+2K(1-2\Sigma)+\phi_n^2K^2}$	1.197	1.108	0.967	0.778	0.689	0.555	0.368	0.120	....
P_n	0.033	0.090	0.185	0.321	0.325	0.319	0.288	0.241	0.284
h_n (ft.) = 151 P_n	5.0	13.6	27.9	48.5	49.1	48.2	43.5	36.4	42.9

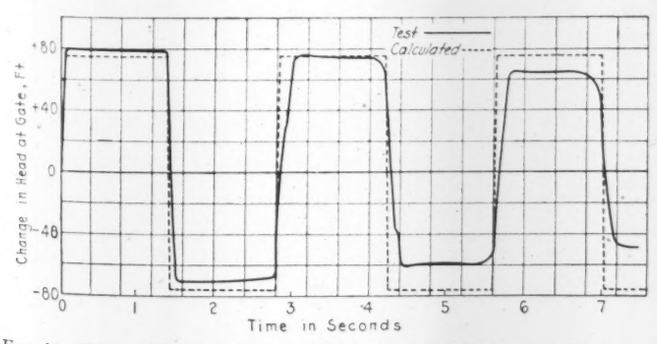

FIG. 11 WATER-HAMMER STUDIES, BIG CREEK EXPERIMENTAL PENSTOCK, SOUTHERN CALIFORNIA EDISON CO.

[Conditions: $H_0 = 301.6$ ft.; $L = 3060$ ft. of 2.06-in. (I. D.) pipe; $2L/a = 1.40$ sec.; $T = 0.04$ sec.; $Q_0 = 0.0127$ cu. ft. per sec. Record at gate.]

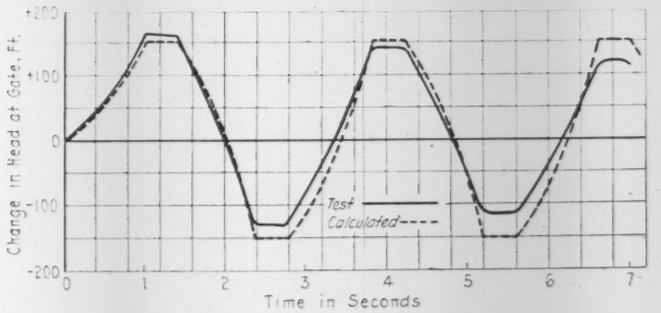

FIG. 12 WATER-HAMMER STUDIES, BIG CREEK EXPERIMENTAL PENSTOCK, SOUTHERN CALIFORNIA EDISON CO.

[Conditions: $H_0 = 301.6$ ft.; $L = 3060$ ft. of 2.06-in. (I. D.) pipe; $2L/a = 1.40$ sec.; $T = 1$ sec. (uniform—see Fig. 10); $Q_0 = 0.0257$ cu. ft. per sec. Record at gate.]

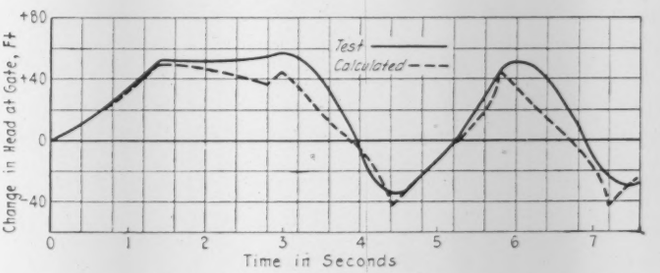

FIG. 13 WATER-HAMMER STUDIES, BIG CREEK EXPERIMENTAL PENSTOCK SOUTHERN CALIFORNIA EDISON CO.

[Conditions: $H_0 = 301.6$ ft.; $L = 3060$ ft. of 2.06-in. (I. D.) pipe; $2L/a = 1.40$ sec., $T = 3$ sec.; $Q_0 = 0.0257$ cu. ft. per sec. Needle travel uniform (see Fig. 10).]

After closure, $P_n = 1 - 2\Sigma$. The wave is symmetrical for subsequent intervals, the ordinates being the same in steps of $2L/a$, but alternately of opposite sign.

n (interval)	←third→			←fourth
t (sec.)	3.30	3.70	4.20	4.40
2Σ	0.818	0.946	1.124	1.282
$P_n = 1-2\Sigma$	0.182	0.054	−0.124	−0.282
h_n (ft.) = 151 P_n	27.5	8.1	−18.7	−42.9

CONCLUSIONS

In conclusion, the elastic-water-column theory provides an accurate method for investigating pressure rise in a closed conduit, for any known gate-closure characteristic. Approximate formulas, though accurate in certain specific cases, should not be used for general problems, as the charts of Figs. 3, 4, and 5 are available for the accurate determination of pressure rise with uniform gate motion and complete closure. The principles of the elastic-water-column theory may be applied as well to the investigation of pressure drop in a closed conduit accompanying gate opening; the work of Allievi,[14] Durand,[15] and Kerr[16] being of particular interest.

The limitations imposed in establishing the characteristics for the ideal pipe line should not materially modify or influence the practical hydraulic-power pipe-line problem. Friction is most always a negligible factor and may be disregarded when small in proportion to the total head. It is difficult to accurately consider its influence as it is distributed along the pipe line. However, Gibson[17] and Durand[15] have discussed a way in which it may be approximated by assuming the friction head as proportional to the square of the velocity in the pipe at the point under consideration. This may be included, in this manner, in Equation [5], and thus introduced into Equation [7].

It has been customary to consider a pipe line of several diameters, thicknesses, etc., of characteristics l_1, A_1, and a_1, as the same hy-

[14] Theory of Water Hammer. Translated by E. E. Halmos, 1925.

[15] Hydraulics of Pipe Lines, 1921.

[16] Fall in Pressure in Hydraulic Turbine Penstocks Due to Acceleration of Flow, N.E.L.A., 1924.

[17] Trans. A.S.C.E., paper No. 1439, vol. 83 (1920), p. 724.

draulically as an equivalent uniform line of characteristics L, A, and a, by making the following substitution:

$$\frac{L}{A} = \Sigma\left[\frac{l_1}{A_1} + \frac{l_2}{A_2} + \cdots\cdots + \frac{l_n}{A_n}\right] = \Sigma\left(\frac{l_n}{A_n}\right)$$

$$\frac{L}{a} = \Sigma\left[\frac{l_1}{a_1} + \frac{l_2}{a_2} + \cdots\cdots + \frac{l_n}{a_n}\right] = \Sigma\left(\frac{l_n}{a_n}\right)$$

thus

$$K = \frac{aV_0}{2gH_0} = \frac{Q_0\Sigma\left(\frac{l_n}{A_n}\right)}{2gH_0\Sigma\left(\frac{l_n}{a_n}\right)}$$

Very few experimental data are available concerning the influence of branch pipes since the tests of Joukovsky.[18] Due to the variety of possible physical characteristics, it is difficult to establish any general conclusions for the purposes of the present paper.

The damping or attenuation of pressure waves should be of little consequence for closure in a few intervals or less, as of interest in hydraulic-power-plant studies. The amount depends upon the pipe line and initial condition. Here again test data must be studied to establish analytical factors as to damping along the pipe line and reflection at the ends.

Acknowledgment is gratefully made to Mr. E. C. Hutchinson,[19] for assistance in the general planning of the work; to Messrs. H. L. Doolittle[20] and R. M. Peabody,[21] for the experimental test data and many valuable suggestions; and to Mr. W. Dreyer[22] for the preparation of the principal charts and continued assistance in the arrangement of the text.

[18] Proc. Am. W. W. Assn., 1904, p. 341.

[19] President and General Manager, The Pelton Water Wheel Co., San Francisco, Cal. Mem. A.S.M.E.

[20] Chief Designing Engineer, The Southern California Edison Co., Los Angeles, Cal. Mem. A.S.M.E.

[21] Assistant Designing Engineer, The Southern California Edison Co., Los Angeles, Cal. Mem. A.S.M.E.

[22] Assistant Engineer, Civil Engineering Department, Pacific Gas & Electric Co., San Francisco, Cal.

The Strength of Pipe Flanges

Approximate Methods of Determining Strength—Proposed Formulas for Strength and Deflection of Rings—Recommended Proportions of Flat Rings—Proposed Method for Determining Strength and Deflection of Hubbed Flanges—Tests on Hubbed Flanges—Recommended Proportions of Hubbed Flanges

By EVERETT O. WATERS[1] AND J. HALL TAYLOR[2]

THE subject of piping for steam, water, and other fluids has attracted an increasing amount of attention of recent years, both from manufacturers and users. Not only have the properties of the various available materials come under review, but the proportions of the pipe itself and the various fittings used in assembling it have been subjected to close study with a view to improvement wherever possible. Higher pressures and temperatures—the familiar demand of the large users of piping—have led to careful studies of piping layouts and design preliminary to actual construction. At the same time there is an insistent call for uniformity and simplification, lest the development of new styles of fittings to meet new conditions should lead to wasteful multiplication of parts carried in stock as "standard."

In the field of flanged fittings much has been accomplished in standardization, but apparently little has been known and less published in regard to the actual strength of the flanges, or as to whether the proportions agreed upon as "standard" are really the best, from the standpoint of strength and economy, for the designated purpose. The American Spiral Pipe Works has therefore undertaken a study of this subject, and the present paper will show (1) the results which have been obtained, as far as flat flanges or rings are concerned, and (2) an extension of the theory to include flanges with hubs, with accompanying tests.

Approximate Methods for Determining Strength

I—Crocker and Sanford, in a leaflet entitled Initial Fibre Stress in Van Stone Flanges, have analyzed the flange as a beam, loaded in one direction by the tension in the bolts, and in the other direction by the compression of the lapped-over end of the pipe. At any axial section of the flange, such as *A-A*, Fig. 1, it is assumed that there is a bending moment induced by that portion of these loads which bears on the flange either to the right or left of the cutting plane, and, according to the simple theory of flexure, this may be equated to the product of the modulus of section *A-A* and the unit stress in the fiber most remote from the neutral axis. The moment due to the bolt load is, according to this theory, equivalent to the moment of a single force concentrated at the center of gravity of the semi-bolt circle, and the moment due to compression from the end of the pipe is likewise taken as the moment of a single force concentrated at the center of gravity of the semi-circular pressure area (shaded in Fig. 1). This is obviously an extension of Bach's approximate method for the strength of flat circular plates supported at the edge.[3] Bach's own criticism of the method—that it presupposes a uniform distribution of the bending moment over the width of the section, and neglects the fact that there are radial as well as hoop stresses—applies with equal force when a solid plate is replaced by a ring.

Such a method has the obvious advantage of simplicity; it uses no complicated formula that must be taken on faith by the person who employs it, but is based on a well-known theory that has met the test of experience in countless structural applications. It also indicates that the stress is a maximum at points where tests have sometimes shown that failure actually does start, i.e., at *S*, Fig. 1. However, as was just stated, certain facts are neglected. For one thing, the only direct stress that is considered is a hoop stress, although a common-sense point of view would indicate that the bending action at any section, such as *ekfghij*, must induce *radial* stresses of tension above and compression below the neutral axis. Moreover the stress is supposed to be uniformly distributed over the width of the section, whereas tests show that the stretching, and hence the stress, differs at the two circumferences of the ring. The theory also presupposes that there is no change in the shape or position of section *A-A* under load, whereas the symmetry of loading indicates that there must be some distortion of this section. That is, if the section *ekfghij* (Fig. 1) bends down

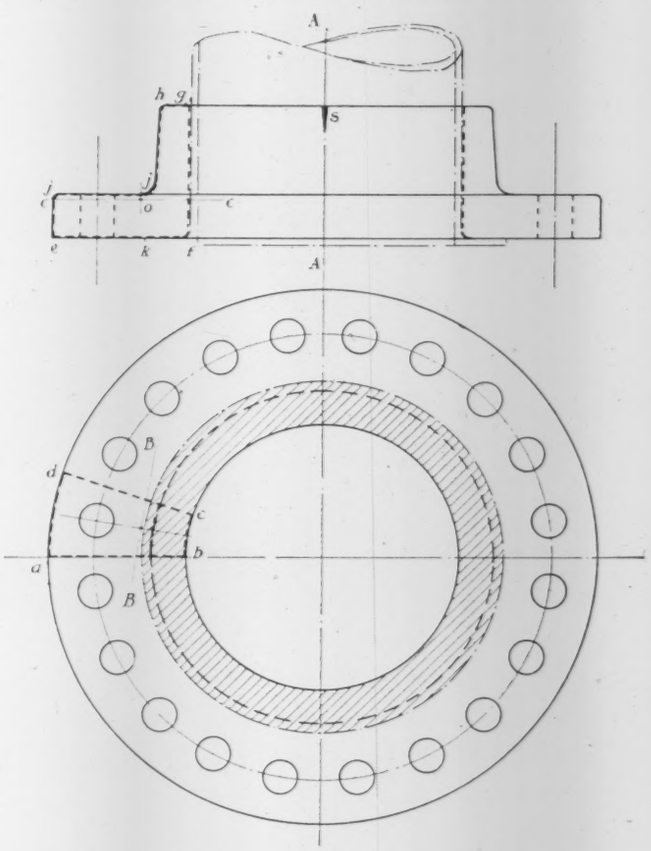

Fig. 1 Plan and Elevation of Flange

under load, the section *A-A* must bend also. Finally, as J. R. Tanner has pointed out, "If a split flange was used there would be no moment of resistance at the split, yet the split flange is used successfully, and frequently under trying conditions."

II—As a substitute for the Crocker-Sanford method, Tanner has proposed that the flange be considered as made up of a number of sectors, and that when loading is applied these sectors remain in position at their outer edges, but are deflected at their inner edges in the direction of the pressure from the pipe end. Each sector is thus a cantilever beam with a depth equal to the thickness of the flange and a length equal to the distance from the bolt circle to the point where pressure is applied, i.e., some arbitrarily chosen point in the shaded area of Fig. 1. The combined width of all the sectors is simply the circumference of the bolt circle.

Here again the simple flexure theory is used, but the conditions, when compared with those of the Crocker-Sanford method, are exactly reversed. The stress is now all radial, and no hoop stress

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[2] President, American Spiral Pipe Works, Chicago, Ill. Mem. A.S.M.E.

[3] C. Bach, Elastizität und Festigkeit, 5th ed., p. 593.

Contributed by the Power Division for presentation at the Spring Meeting, White Sulphur Springs, W. Va., May 23 to 26, 1927, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

Be sure to bring this copy with you to the Spring Meeting

whatever is considered. Obviously, the method does not work when hubbed flanges are used. Crocker has suggested that this difficulty can be overcome by assuming that the sectors are fixed at their inner ends—which is to a large extent true, due to the very powerful strengthening effect of the hub—and deflect at the outer edge. The net effect of this modification is to substitute the circumference of the hub at the outside (or the raised male if that is greater) for the circumference of the bolt circle.

III—A third method appeared in *The Locomotive* of July, 1905, published by the Hartford Steam Boiler Inspection and Insurance Company. This article, while it purported to treat only of mouth-

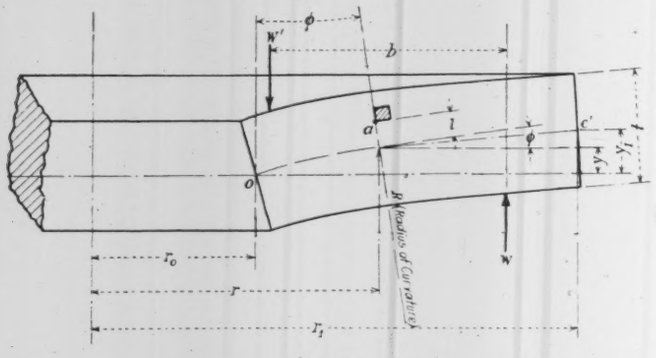


FIG. 2

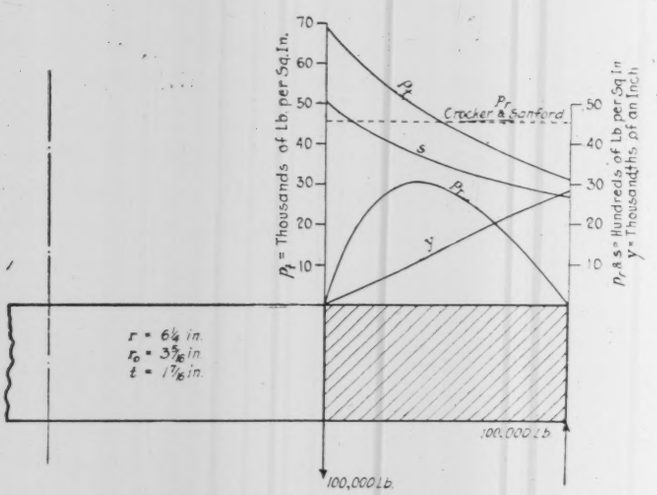


FIG. 3 STRESSES IN STEEL RING

piece rings of vulcanizers and similar vessels, applies equally well to pipe flanges of any shape. The analysis is briefly as follows: The flange is divided into sectors, with one bolt hole per sector (see heavy dotted outline *abcd*, Fig. 1). The forces acting on the sector are (1) the bolt pull, (2) the compression of the pipe end, and (3) normal stresses of varying magnitude at the two faces *ab* and *dc;* and, since the sector is in equilibrium, the sum of the moments of the components of all these forces equals zero, when taken about any axis whatever. By making one or two assumptions, such as that the section *efghij* does not change shape under load, it is proved that the normal stresses at the faces *ab* and *cd* are proportional to their distance from the "neutral plane" *C-C* through the center of gravity *O*, just as in a beam subjected to bending. Therefore, by taking for axis of moments the line *B-B* through the center of gravity *O*, an equation is set up between the "external" moments due to bolt pull and pipe pressure on the one hand, and the "internal" moment due to the varying stresses at *ab* and *cd* on the other. From this equation, which is analogous to the flexure formula for beams, the hoop stress may be calculated for any point in the flange.

It will be noticed that this method has one point of superiority, in that it recognizes that the deformation of the flange under load is in the nature of a dishing rather than bending; but it assumes a rigidity of cross-section and absence of radial stress, both of which are contrary to fact.

As might be expected, the three methods do not give the same results when applied to the same problem. For example, assume that a 14-in. flange for a Van Stone joint has the following dimensions (see Fig. 1):[4]

$ef = 4^3/_8$ in. $fg = 5^3/_8$ in.
$ej = 2^1/_8$ in. $hg = 1^1/_4$ in.
Diameter to $k = 17^3/_8$ in.

Bolt-circle diameter = $20^1/_4$ in. Number of bolt holes = 20. Bore = $14^1/_4$ in. Bolt-hole diameter = $1^1/_4$ in. Load on each bolt = 6940 lb.

Then according to the Crocker-Sanford method,

$$\text{Bending moment} = 6940 \times \frac{20}{2}\left[\frac{20.25}{\pi} - \frac{2(17.375^3 - 14.25^3)}{3\pi(17.375^2 - 14.25^2)}\right]$$
$$= 97{,}100 \text{ in-lb.}$$

Section modulus (at section through bolt holes) = 14.75 in.^{3}
Maximum stress = 6580 lb. per sq. in.

According to the Tanner method (modified as suggested by Crocker)

$$\text{Bending moment} = 6940 \times 20\left(\frac{20.25 - 17.375}{2}\right) = 199{,}500 \text{ in-lb.}$$

Section modulus (at outer circumference of pipe end, k) = 41.1 in.^{3}
Maximum stress = 4860 lb. per sq. in.

According to the method published in *The Locomotive*,

Moment of external forces per "section" =

$$6940\left(\frac{20.25 - \left(\frac{17.375 + 14.25}{2}\right)}{2}\right) = 15{,}400 \text{ in-lb.}$$

Section modulus (at section through bolt hole) $= \frac{2\pi}{20} \times \frac{14.75}{2}$
$= 2.32$ in.^{3}

Maximum stress = 6630 lb. per sq. in.

The stress by the Tanner method is low, evidently because the extreme fibers of the high hub are not brought into consideration; the other two methods agree very closely in their results—in fact, the detailed calculations show that they are practically identical; although they approach the solution from totally different angles, the approximations and adjustments in each case are such as to bring them into line with each other.

After the foregoing review, it seems clear that a better analysis of flange stresses is demanded which will (1) give due consideration to both radial and hoop stresses, and (2) incorporate a load-deflection relation as well as a load-stress relation, so that tests can be checked and the formulas verified. Presumably, if the load-deflection relation agrees with load-deflection results given by the testing machine, the load-stress relation will be correct. With this premise, the classical method of Winkler and Grashof[5] for flat circular plates has been adapted to the problem in hand, and some very interesting results obtained. The solution for flat rings is first submitted, after which are given the results of tests and the modification for hubbed flanges.

PROPOSED FORMULAS FOR STRENGTH AND DEFLECTION OF RINGS

For the sake of simplicity, the following approximations[6] are made which may be corrected later, if desired, by supplementary analysis:

1 The bolt load is distributed uniformly over the circumference of the bolt circle.

2 The proportions of all commercial rings are such that the

[4] This size of flange was used as an illustration of the Crocker-Sanford method in the pamphlet by those authors, previously cited.

[5] See Bach, Elastizität und Festigkeit, p. 567. Föppl (Vorlesungen über Technische Mechanik, III, p. 273) follows a different treatment, but reaches identical results for the special cases that are worked out completely. See also Timoshenko and Lessells, Applied Elasticity, p. 260, for a treatment of the subject in English.

[6] In addition to those assumed by Grashof, e.g., straight lines perpendicular to the meridian section before flexure remain straight and perpendicular after flexure, and stretching of the meridian section is neglected.

bolt load W (Fig. 2) is applied near the outer edge and the pipe load W' is applied at or near the inner edge; hence the stresses and deflections are sensibly equal to those that would be produced by loads P and P' at the outer and inner edges, respectively, P and P' being equal to $Wb/(r_1 - r_0)$. In other words, the principle of *equivalent couples*, Wb and $P(r_1 - r_0)$, is used.

The following symbols are used in solving the problem: (see Figs. 2 and 20).

- b = radial distance from bolt circle to center of pressure of pipe end, in.
- E = Young's modulus, lb. per sq. in.
- e_h = axial strain, in hub portion of flange
- e_r = radial strain, in plain ring or ring portion of flange
- e_t = tangential strain, in plain ring or ring portion of flange
- e_t' = tangential strain, in hub portion of flange
- f = radial distance from neutral surface of hub to random point a', in.
- g = thickness of hub, in.
- h = distance from end of hub to random point a', in.
- h_1 = distance from end of hub to meridian section of ring, in.
- i = radial distance from neutral surface of hub to outside of hub, in.
- K = r/r_0
- K_1 = r_1/r_0
- l = perpendicular distance from meridian section to random point a
- M = loading moment, in-lb.
  = $P(r_1 - r_0)$
  = $Pr_0(K_1 - 1)$
- **M** = moment of axial stresses in hub, for a sector of unit circumferential length, lb-in. per in.
- M_0 = moment of radial stresses at junction of hub and flange, acting on a sector of unit length, at the junction, lb-in. per in.
- $1/m$ = Poisson's ratio (0.3 for steel, 0.25 for cast iron)
- n = slope of meridian line at point c (Fig. 2) or O (Fig. 20)
- P = effective load, lb.
- p_h = axial stress, lb. per sq. in., in hub portion of flange
- p_r = radial stress, lb. per sq. in.
- p_{r0} = radial stress, lb. per sq. in., at the virtual inside edge of ring or flange, where $r = r_0$ and $l = t/2$
- p_t = tangential stress, lb. per sq. in., in plain ring or ring portion of flange
- p_t' = tangential stress, lb. per sq. in., in hub portion of flange
- p_{t0} = tangential stress, lb. per sq. in., at the virtual inside edges of ring or flange, where $r = r_0$ and $l = t/2$
- R = radius of curvature of meridian line CC', in.
- r = radius to intersection of meridian surface of ring with normal cone containing random point a, in.
- r_1 = outside radius of ring, in.
- r_0 = inside radius of ring, or virtual inside edge of hubbed flange, in.
- s = shear, lb. per sq. in.
- t = thickness of ring, in.
- W = W' = actual bolt load, lb.
- x = radius to neutral surface of hub, in plane normal to axis and containing random point a', in.
- y = deflection of ring, measured at the meridian section at radius r, in.
- y_1 = total deflection of ring, measured at the meridian section, in.

1 The tangential strain at a random point a is

$$e_t = \frac{2\pi\left(r - l\frac{dy}{dr}\right) - 2\pi r}{2\pi r} = -\frac{l}{r}\frac{dy}{dr};$$

the radial strain is

$$e_r = \frac{\frac{R+l}{R}(r_1 - r) - (r_1 - r)}{r_1 - r} = \frac{l}{R} = -l\frac{d^2y}{dr^2};$$

hence the expressions for tangential and radial stresses at a are, from the theory of elasticity:

$$p_t = \frac{mE}{m^2 - 1}(me_t + e_r) = -\frac{mEl}{m^2 - 1}\left(\frac{m}{r}\frac{dy}{dr} + \frac{d^2y}{dr^2}\right) \text{ and}$$

$$p_r = \frac{mE}{m^2 - 1}(me_r + e_t) = -\frac{mEl}{m^2 - 1}\left(m\frac{d^2y}{dr^2} + \frac{1}{r}\frac{dy}{dr}\right).$$

2 Consider the loads acting in a radial direction on the small element at a, whose height is dl, radial length is dr, and tangential length is $rd\theta$; their sum, equated to zero, and neglecting products of differentials is

$$\frac{dp_r}{dr} + \frac{ds}{dl} + \frac{p_r - p_t}{r} = 0$$

s being the unit shear at point a.

3 Substituting the values of p_t and p_r in this equation gives

$$ds = \frac{m^2E}{m^2 - 1}\left(\frac{d^3y}{dr^3} + \frac{1}{r}\frac{d^2y}{dr^2} - \frac{1}{r^2}\frac{dy}{dr}\right)ldl$$

which when integrated gives

$$s = \frac{m^2E}{m^2 - 1}\left(\frac{d^3y}{dr^3} + \frac{1}{r}\frac{d^2y}{dr^2} - \frac{1}{r^2}\frac{dy}{dr}\right)\frac{l^2}{2} + A$$

At the surface of the ring where $l = \pm t/2$, the shear is zero. Evaluating the constant A by means of this fact, the equation for the unit shear at any point in the ring is

$$s = \frac{m^2E}{8(m^2 - 1)}\left(\frac{d^3y}{dr^3} + \frac{1}{r}\frac{d^2y}{dr^2} - \frac{1}{r^2}\frac{dy}{dr}\right)(4l^2 - t^2)$$

4 The total shearing force over the conical section of the ring which contains point a, is the integral of $2\pi rsdl$ between the limits $t/2$ and $-t/2$, and this is also equal to P; hence there is obtained for the general equation of the meridian line CC':

$$\frac{d^3y}{dr^3} + \frac{1}{r}\frac{d^2y}{dr^2} - \frac{1}{r^2}\frac{dy}{dr} = -\frac{6(m^2 - 1)P}{\pi m^2Et^3r}$$

5 In the process of solving this equation, three constants of integration appear. These are evaluated by the fact that $y = 0$ when $r = r_0$, $p_r = 0$ when $r = r_1$, and $p_r = 0$ when $r = r_0$. The final equation for the elastic curve CC' is then obtained. By successive differentiation, and substitution in the formulas for p_t, p_r, and s, equations are obtained for the stresses and shear at any point, in terms of the load P, the radial distance r to the point, and the perpendicular distance l, from the meridian section to the point.

These equations are rather lengthy and are of no particular importance here, but are included in an appendix for reference purposes. Attention is called to Fig. 3, in which the meridian line, normal stresses at the surface, and shear at the meridian section, have been plotted to scale for a steel ring of the dimensions indicated, under a loading of 100,000 lb. at the rims. Young's modulus is taken as 30,000,000 lb. per sq. in., and Poisson's ratio as 0.3. These curves indicate that:

1 The deflected surface of a ring of commercial proportions is very nearly a true cone. If the outside diameter is much greater than the inside, this is not true, and calculations show that each surface has both convex and concave curvature with lines of inflection.

2 The radial stress is comparatively small and does not appreciably affect the strength of the ring.

3 The maximum shear and maximum tangential stress occur at the inner edge, and fall off rapidly toward the outer edge, where the tangential stress has only about one half its value at the inner edge.

4 The tangential stress at the inner edge is the real determining factor in the strength of the ring.

The results that are of most value are the formulas for the maximum stress (p_t at the inner edge) and the total deflection (y when $r = r_1$). They are, when E = 30,000,000 lb. per sq. in. and m = 10/3:

$$p_{t0} = \frac{2.858M}{t^2 r_0 (K_1 - 1)} \left(\frac{K_1^2 \log_{10} K_1}{K_1^2 - 1} + 0.1169 \right) \ldots\ldots [1]$$

$$y_1 = \frac{0.000000285 M r_0}{t^3 (K_1 - 1)} \left[\frac{K_1^2 (\log_{10} K_1)^2}{K_1^2 - 1} + 0.0644 (K_1^2 - 1) \right] \ldots [2]$$

VALIDITY OF PROPOSED FORMULAS

No theory is worth the effort required to develop it if it does not bear the test of experiment. In the present instance some rather bold assumptions were made which could hardly expect to pass unchallenged, and it would be folly to claim that the theory is perfect. It might even be doubted whether it has any superiority over the other theories mentioned in earlier paragraphs, were it not for the fact that formula [2] checks strikingly with the results of tests that have recently been made on flat rings. *Inasmuch as the deflections obtained in these tests agree with those calculated from formula [2], it may be assumed that the theory underlying both formulas [1] and [2] is nearly enough correct for practical purposes, and that formula [1] will give the maximum stress that actually does occur in the rings.*

The comparison between theory and experiment is shown in Figs. 4–16. The first six (Figs. 4–9) tests were reported by J. R. Tanner to the Working Committee of Sub-Committee No. 3 on the Standardization of Pipe Flanges and Fittings, in November, 1923. In the set-up for these tests, load was applied to the inner edges of the rings by a "supporting ring," whose outside diameter corresponded to the outside diameter of the small tongues that would be standard for the pipe sizes in question; the opposing load was applied at the bolt circle to "bearing beads" or buttons inserted in the bolt holes. The deflections were measured in such a way as to include the crushing of the spherical faces of the "bearing beads" and the bearing surfaces of the "supporting ring," and also

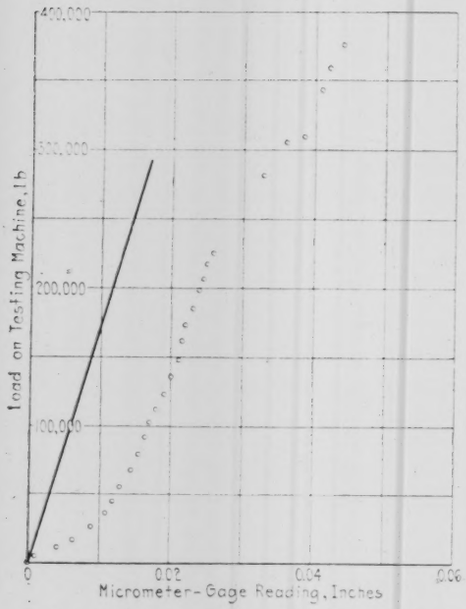


Fig. 4—Test No. 1. Ring 8⅝ in. I. D., 18¹⁵/₃₂ in. O. D., 2⅜ in. thick.

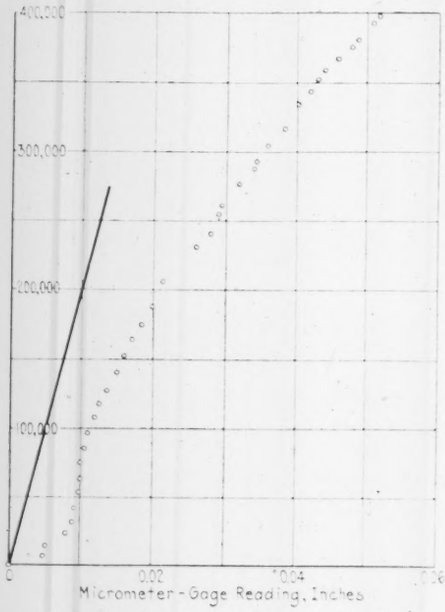


Fig. 5—Test No. 3. Ring 8⅝ in. I. D., 16½ in. O. D., 2½ in. thick.

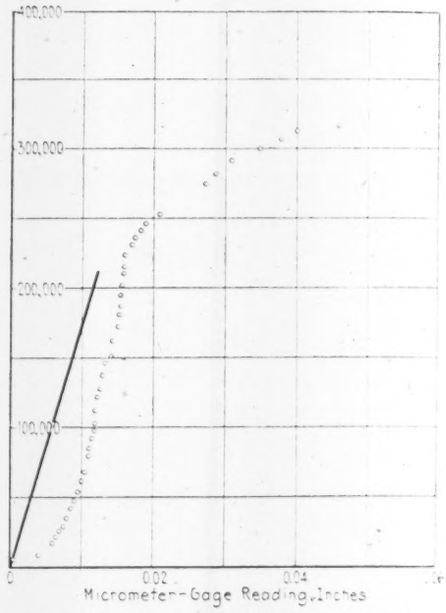


Fig. 6—Test No. 4. Ring 8⅝ in. I. D., 15 in. O. D., 1⅞ in. thick.[1]

[1] Thickness not tabulated in report of Nov. 19, 1923, to the Working Committee of Sub-Committee No. 3. The value of 1⅞ in. was assumed on the strength of a statement in the report that the flange of Test No. 4 was a proposed 400-lb. standard.

FIGS. 4–6 TESTS OF PLAIN RINGS, PITTSBURGH VALVE, FOUNDRY AND CONSTRUCTION CO., 1923—STANDARD DRILLING FOR BOLT HOLES

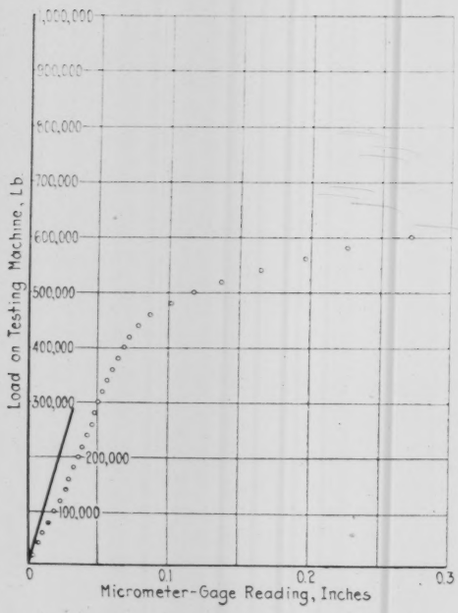


Fig. 7—Test No. 2. Ring 16 in. I. D., 25¹⁵/₃₂ in. O. D., 2¹/₈ in. thick.

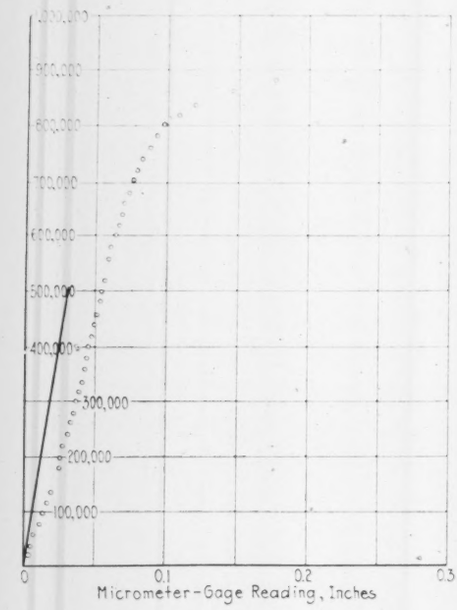


Fig. 8—Test No. 5. Ring 16 in. I. D., 26¹⁵/₁₆ in. O. D., 2⅞ in. thick.

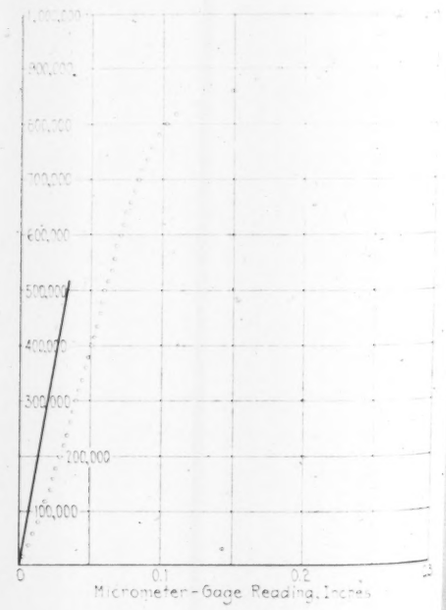


Fig. 9—Test No. 6. Ring 16 in. I. D., 27¼ in. O. D., 3¹/₆₄ in. thick.

FIGS. 7–9 TESTS OF PLAIN RINGS, PITTSBURGH VALVE, FOUNDRY AND CONSTRUCTION CO., 1923—STANDARD DRILLING FOR BOLT HOLES

in the case of the 16-in. rings, included the spring of various parts of the testing machine itself. For these reasons the test points do not coincide with the calculated line, but, when they once straighten out, a very close parallelism is indicated, proving the validity of the theory as far as stress-strain relations are concerned. In plotting the lines, formula [2] was used to obtain the theoretical deflection at the outside diameter; this was then reduced in the ratio $b/(r_1 - r_0)$ (see Fig. 2) to obtain the deflection at the bolt circle, on the assumption that y is proportional to $r - r_0$.

Realizing that comparatively little experimenting had hitherto been done and that there was much to be learned about the subject, the American Spiral Pipe Works conducted a series of tests in July, 1924, under Mr. Taylor's supervision, on 6-in. flanges with the drilling and outside diameter the same as the 250-lb. standard. In an attempt to make the conditions of the tests as close as possible to those existing in the field, the rig shown in Fig. 17 was made up; in this arrangement the flange was supported by alloy-steel bolts, the nuts at the lower end engaging the back of the flange and those at the upper end bearing on a ring which received its support from the lower platen of the testing machine. The pressure ram with a slot across the bottom for the gage bars was placed on top of the flange and engaged the face of the latter about the same as a gasket when in actual use.

Special care was taken to make the strain as nearly as possible equal on all bolts; this was done after putting a load of 10,000 lb. on the pressure ram. A micrometer gage was attached to the lower gage bar, which rested on the face of the flange at points close to the inside edge; another gage bar, placed above this, engaged the face of the flange at the outer rim, and carried a pin which operated the micrometer, thus indicating the deflection of the flange from the original plane of the face, between the outer rim and the points where the inner gage bar rested. The flanges were loaded and deflections noted at intervals of 20,000 lb.; each reading was taken twice and the load was reduced to zero each time, before putting on the next increment.

It will be noted that one source of error in the previous tests has been eliminated, by using gage bars that have no connection with any part of the testing machine and therefore indicate correctly the deflection of the test specimen and nothing else. Figs. 10–12 show the results of these tests, as far as the flat rings are concerned, and it will be seen that the observations are consistent with the theory.

In order to furnish further data on the subject of stresses in flanges, and check the formulas being developed by Mr. Waters, a new series of tests was made during July, 1926. Fig. 18 shows the set-up and method employed in running the tests. The flange was supported at each bolt hole by hardened steel plugs, the bearing point of each plug being a 1 1/2-in. steel ball. This allowed the flange to be supported uniformly on a series of points each of which was exactly on the bolt circle, and was a decided improvement over the bolt supports that were used in the 1924 series, inasmuch as all possible constraint was removed from the free deflection of the flange under load. To facilitate rapid change of the flanges in the testing machine each pressure plug with its ball and support was enclosed in a light sleeve made of several thicknesses of tin plate; this held the parts in alignment, yet allowed for deflection. Loading was applied to the face of the flange through a pressure ram similar to that used in the previous tests. An initial load of 4000 lb. was first applied, to assure a firm setting on which to base the reading; the loads were then increased in steps of 20,000 lb. and deflection readings taken at each increase. After each load was applied, a reading was taken at 4000 lb. to determine the permanent set.

Rings 1–6 of this series were tested with full-length outside gage bar, as indicated in Fig. 18 (*a*); for rings 7 and 8 this was changed as shown in Fig. 18 (*b*), where the gage bar is shown resting on 1/8-in. × 1/8-in. × 1 1/2-in. keys whose outer edges are flush with the bolt circle. This gave a direct reading of the deflection at the bolt circle. However, inasmuch as both calculation and micrometer measurements of the deformed surfaces of tested rings showed that these surfaces are to all intents and purposes perfect cones, it is not believed that this change makes any appreciable difference in the value of the final results. In all cases (Figs. 13–16) the calculated line has been adjusted to indicate the deflection at the same points at which the gage bars were applied, using the relation $y/(r - r_0) = \text{constant}$.

In the 1926 tests an effort was made to ascertain what effect, if any, the bolt holes had upon the strength of the rings. Rings were prepared without standard bolt holes, but drilled with 1/4-in. holes merely to line up the centering plugs, and these were tested in comparison with other rings of the same dimensions and having standard bolt-hole drilling. Apparently there is no marked difference in strength at loadings within the limit of proportionality, but above this point the rings with full-sized holes showed a tendency to fail at lower loads than those with the small holes. This can be explained on rational grounds, remembering that in the theoretical ring, without holes and loaded at the inner and outer edges, the stress is a maximum at the inner edge and falls off rapidly toward the outer edge. Such a ring, when stressed above the elastic limit, changes its rate of deflection very slowly, because comparatively few fibers are overstressed; the full-size bolt holes, on the other hand, raise the stress at the bolt circle to a value comparable with that at the inner edge, so that, when the elastic limit is passed, a larger number of fibers are overstressed and a sharper change in the rate of deflection takes place. In other words, the transition from elastic to plastic deformation is more definite and complete.

In explanation of the charts, it should be stated that the straight lines have been made to terminate at the load which gives, by computation, the yield point that was determined for the ring in question by a separate tensile test on a sample coupon taken from the same billet. The straight lines should therefore end, if the theory is correct, and neglecting any change in elastic limit due to working the material from billet to final product, at the same load at which the plotted points begin to show a deviation from the proportionality law. For the 1926 tests the value of E was taken from the tensile tests on the sample coupons and ranged from about 27,000,000 to 32,000,000 lb. per sq. in.; in the remaining cases no specific data were available and E was assumed as 30,000,000 lb. per sq. in.

After the flanges were tested a straight edge was placed across the face in order to determine its shape and it was found to be a perfectly straight line from the outside to the inside edge. But to further check this in a more accurate manner, the flanges were set up as shown in Fig. 19. A surface gage was used and the flange was tipped up on one side by means of a set screw and nut until the surface gage showed no variation between the inside and outside edge of the face. The surface gage was then run in a straight line over the face and readings taken every quarter of an inch. The gage did not show any appreciable variation and thus proved the surface of the flange to be dished in a straight line from the inside to the outside edge.

Recommended Proportions of Flat Rings

The determining factors in proportioning rings for piping and pressure vessels are, first, outside diameter of pipe, and second, fluid pressure within the pipe. Furthermore, examination of the stress formula [1] on the preceding page shows that economy of material is secured by using thick, narrow rings rather than thin, wide rings. Therefore it is recommended that the following method be used in designing rings, wherever existing standards do not already apply:

1 Select arbitrarily a bolt diameter, and determine the number of bolts such that the total safe load equals the sum of the load due to the fluid pressure and that due to the desired gasket pressure. Ordinarily the number of bolts will be a multiple of 4.

2 Determine the bolt-circle diameter, such that (*a*) bolt holes are at least 2 1/4 diameters apart, measured center to center; and (*b*) ample nut clearance is provided, for use of socket or open-end wrenches. The first of these provisions will prevent excessive stress concentration between the bolt holes.

3 Determine the outside diameter of the ring, making it no larger than necessary to secure full seating of the bolt heads and nuts.

4 Determine the ring thickness from Equation [1], using the working strength of the bolts to obtain the moment M, and the working strength of the ring material for p_w.

Any pair of rings designed in this manner will be economical of

material, and at least as strong as the bolts holding them together.

PROPOSED METHOD FOR DETERMINING STRENGTH AND DEFLECTION OF HUBBED FLANGES

Consider the flange of Fig. 20. It consists of a ring of cross-section *ABCD*, and a short tube of cross-section *AEFG*. It may be assumed that the ring is loaded by a force P at the outer edge, an equal and opposite force P at the neutral axis of the tube (which provided we assume that a numerical value for n can be found.

The analysis for plain rings has also shown that at any given distance from the axis of the ring the radial and tangential stresses are zero at the meridian surface, and vary directly as their distance from that surface. The moment of the radial stresses, acting on a sector of a ring whose circumferential length is equal to unity at the place where the stresses occur, is therefore $[t^2/6 \times p_r]_{t=t/2}$. At point O, Fig. 20, this is the previously mentioned moment

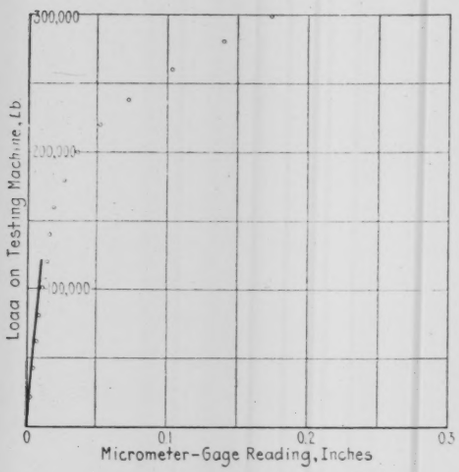


Fig. 10—Test No. 1-A, 1924. Ring 6⅝ in. I. D., 12⅛ in. O. D., 1⅝ in. thick, standard drilling for bolt holes.

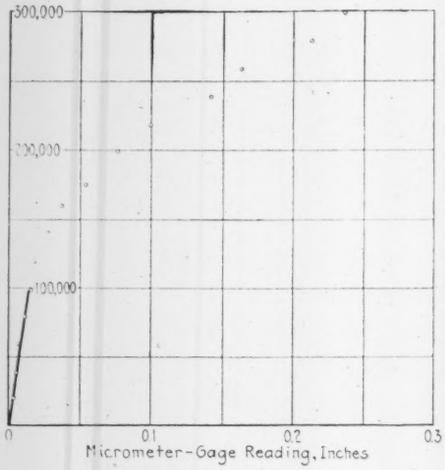


Fig. 11—Test No. 2-A, 1924. Ring 6⅝ in. I. D., 12½ in. O. D., 1⅞ in. thick, standard drilling for bolt holes.

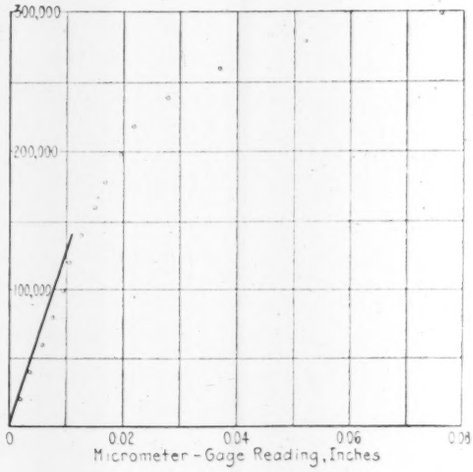


Fig. 12—Test No. 2-B, 1924. Ring 6⅝ in. I. D., 12½ in. O. D., 1¾ in. thick, standard drilling for bolt holes.

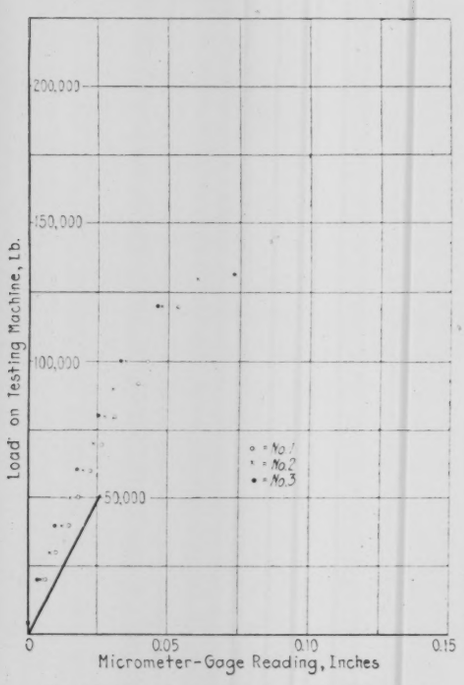


Fig. 13—Test rings Nos. 1, 2, and 3, 1926. Rings 14¼ in. I. D., 18 in. O. D., 15/16 in. thick. Nos. 1 and 2, 28 ¼-in. holes on standard bolt circle; No. 3, standard drilling for bolt holes.

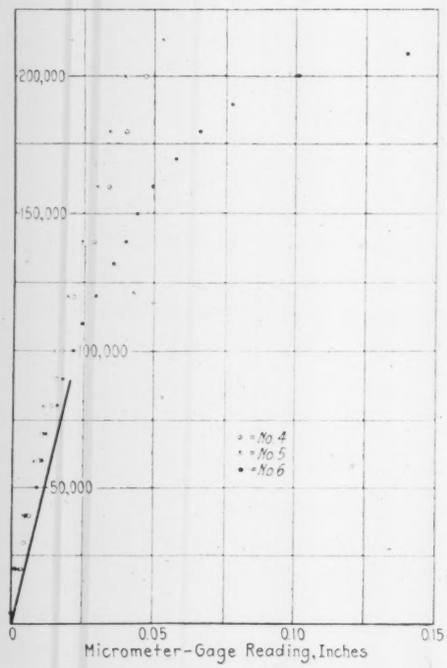


Fig. 14—Test rings Nos. 4, 5, and 6, 1926. Rings 14½ in. I. D., 18 in. O. D., 1¼ in. thick. Nos. 4 and 5, 28 ¼-in. holes on standard bolt circle; No. 6, standard drilling for bolt holes.

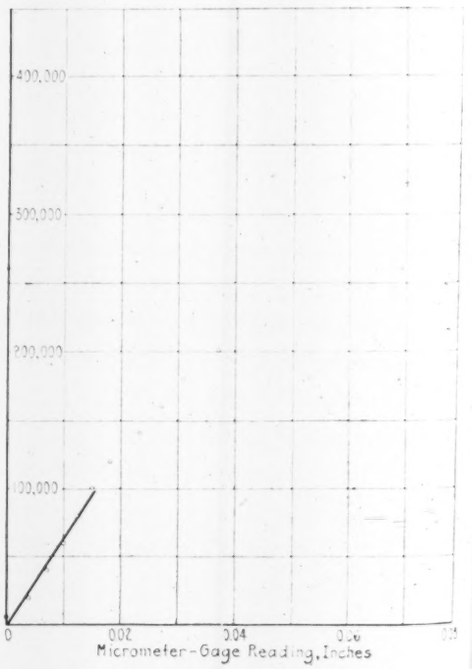


Fig. 15—Test ring No. 7, 1926. Ring 10¾ in. I. D., 17½ in. O. D., 1½ in. thick, standard drilling for bolt holes.

FIGS. 10–15 TESTS OF PLAIN RINGS, AMERICAN SPIRAL PIPE WORKS, 1924–1926

is the virtual inner edge), and a moment at O tending to resist dishing of the ring. The tube, on the other hand, is subjected to no external force, but withstands an equal and opposite moment at O, tending to crimp it into the form shown in the figure.

The radial and tangential stresses p_r and p_t and the shear s in the ring *ABCD* are related by the same general equations as before and the differential equation for the meridian line is unchanged. To solve this equation, three constants of integration must be evaluated. Two of these are obtained from the fact that $y = 0$ when $r = r_0$, and $p_r = 0$ when $r = r_1$. The third may be obtained from an arbitrary equation, $dy/dr = n$ when $r = r_0$, acting on the ring portion of the flange, whose section is *ABCD*. We therefore have the relation $\mathbf{M}_0 = p_{r0}t^2/6$, where $\mathbf{M}_0$ is the external moment acting on a sector of the ring portion whose circumferential length is unity at point O. Since p_{r0} can be evaluated in terms of particular values of y and r, and the unknown slope n, this equation gives one relation between $\mathbf{M}_0$ and n. It remains to find an additional relation between $\mathbf{M}_0$ and n, in terms of known quantities, in order to get numerical solutions for $\mathbf{M}_0$ and n and hence for the equations for y, p_r, and p_t. This is done by analyzing the stress conditions in the tube or hub portion, in the following manner:

1 The tangential strain at a random point a', Fig. 20, is $e_t' = (x - r_0)/r_0$, assuming that the thickness of the hub is small in comparison with the radius; the axial strain is $e_h = -fd^2x/dh^2$; hence the expressions for tangential and axial stresses at a' are:

$$p_t' = \frac{mE}{m^2-1}(me_t' + e_h) = \frac{mE}{m^2-1}\left[\frac{m}{r_0}(x - r_0) - f\frac{d^2x}{dh^2}\right] \text{ and}$$

$$p_h = \frac{mE}{m^2-1}(me_h + e_t') = \frac{mE}{m^2-1}\left[-mf\frac{d^2x}{dh^2} + \frac{x - r_0}{r_0}\right]$$

2 Consider the cone whose axis is the axis of the flange, and whose surface contains point a' and is normal to the inner and outer

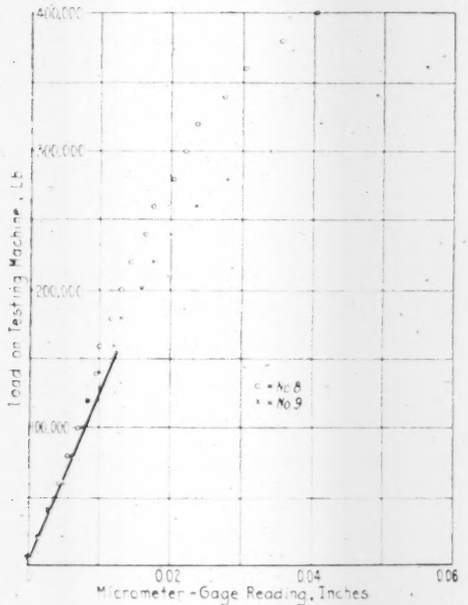


FIG. 16 TESTS OF PLAIN RINGS, AMERICAN SPIRAL PIPE WORKS, 1924–1926

Test rings Nos. 8 and 9, 1926. Rings 10¾ in. I. D., 17½ in. O. D., 1⅞ in. thick. No. 8, 16 ¼-in. holes on standard bolt circle; No. 9, standard drilling for bolt holes.

surfaces of the hub at the lines of intersection. The axial stresses across the conical surface are partly tensile, partly compressive; but their sum is zero. From this fact may be derived an expression for the location of the neutral surface of the hub:

$$i = \frac{g}{2} + \frac{x - r_0}{mr_0\dfrac{d^2x}{dh^2}}$$

3 The moment of the above axial stresses, for a sector of the hub of unit circumferential length, taken about the neutral surface, is

$$\mathbf{M} = \int_{i-g}^{i} p_x f df = -\frac{m^2Eg^3}{12(m^2-1)}\frac{d^2x}{dh^2}$$

4 The tangential force acting on one of the faces of the above sector, of unit length in the axial direction, is

$$\Sigma p_t' = \int_{i-g}^{i} p_t' df = \frac{Eg(x - r_0)}{r_0}$$

and the entire tangential force, from a' to the lower end of the hub, is

$$\int_h^0 \frac{Eg(x - r_0)dh}{r_0}$$

5 This sector of the hub, of unit circumferential length and having a height h, is in equilibrium under the action of two tangential forces of the above value each, and a horizontal shear at the upper end: there are no other forces having *horizontal* components. And the faces of this sector at which the tangential forces act, make an angle $1/r_0$, so small that sin $1/2r_0$ sensibly equals $1/2r_0$. Hence is derived the equilibrium equation

$$\frac{2}{2r_0}\int_h^0 \frac{Eg(x - r_0)dh}{r_0} = \text{total horizontal shear at the surface through } a'.$$

6 The above shear, as is well known, equals the first derivative, with respect to h, of the moment at the surface through a'; therefore

$$\frac{2}{2r_0}\int_h^0 \frac{Eg(x - r_0)dh}{r_0} = \frac{d\mathbf{M}}{dh} = -\frac{m^2Eg^3}{12(m^2-1)}\frac{d^3x}{dh^3}$$

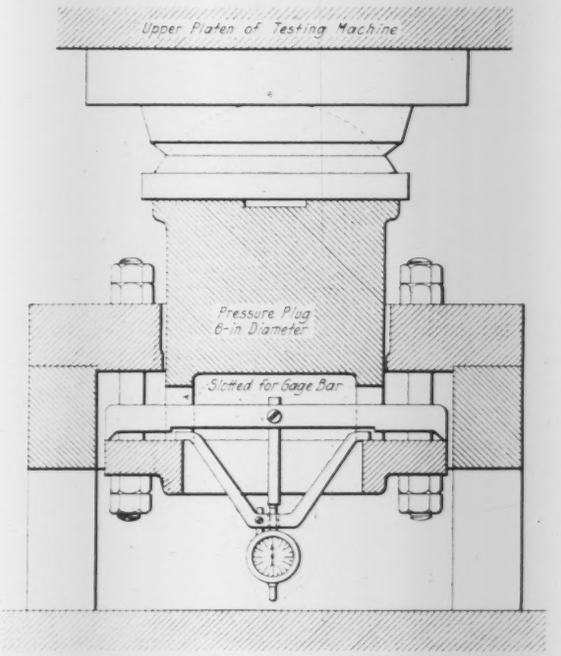


FIG. 17 RIG USED IN 1924 TESTS OF FLANGES

By differentiating and simplifying, we obtain the general equation for the elastic curve of the hub.[7]

$$\frac{d^4x}{dh^4} + \frac{12(m^2-1)(x - r_0)}{(mgr_0)^2} = 0$$

This requires the evaluation of four constants, for complete solution. The fact that $x - r_0 = 0$, when $h = h_1$, the moment $\mu = 0$ when $h = 0$, the shear $= 0$ when $h = 0$, and the slope $dx/dh = n$ when $h = h_1$, satisfies this condition, again with the proviso that a numerical value for n can be obtained. However, it is also true that $\mathbf{M} = \mathbf{M}_0$ when $h = h_1$; in other words,

$$\mathbf{M}_0 = \left[-\frac{m^2Eg^3}{12(m^2-1)}\frac{d^2x}{dh^2}\right]_{h = h_1}$$

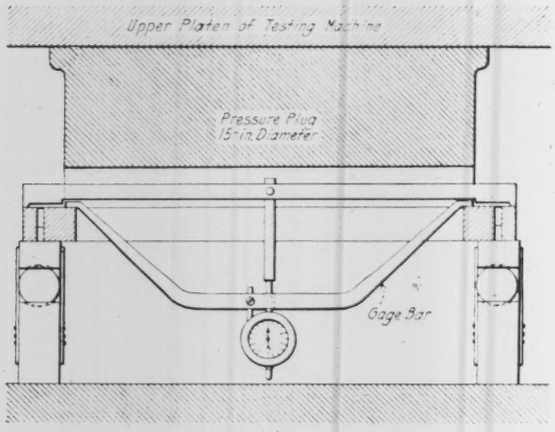
an equation solvable, for particular values of x and h, in terms of n. This gives the second of the two necessary relations between $\mathbf{M}_0$ and n, and the whole structure of equations may therefore be solved with numerical answers.

The most important results are those that give the principal stresses at certain definite points, and the deflection of the ring portion of the flange, in terms of the load and the known dimensions of the flange. These are as follows, when, as previously, E = 30,000,000 lb. per sq. in. and $m = 10/3$:

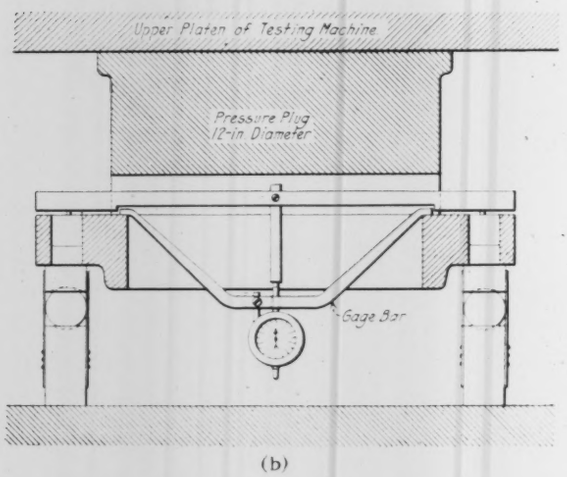
For the tangential stress at b, Fig. 20.

[7] The same type of equation is obtained for a beam loaded at a point and resting over its entire length on an elastic support—a somewhat similar case. See Timoshenko and Lessells: Applied Elasticity, p. 133.

$$p_{r0} = \frac{2.86M}{r_0(K_1 - 1)t^2}\left\{\frac{K_1^2 \log_{10} K_1}{K_1^2 - 1} + 0.177 - \frac{K_1^2 + 1}{K_1^2 - 1}\left[\frac{3.33K_1^2 \log_{10} K_1 + 0.389\,(K_1^2 - 1)}{4.33K_1^2 + 2.33 + 1.18\,\dfrac{(K_1^2 - 1)t^3D}{\sqrt{r_0g^5}}}\right]\right\} \quad \ldots [3]$$



(a)



(b)

FIG. 18 RIG USED IN 1926 TESTS OF FLANGES

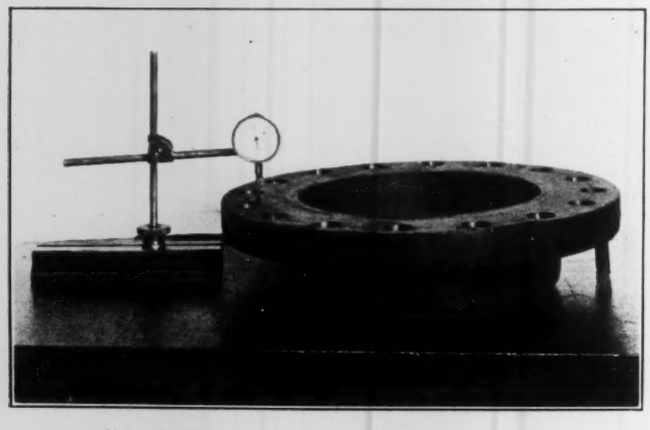


FIG. 19 TESTING FLANGES FOR DISHING OF FACE

For the radial stress at b,

$$p_{r0} = \frac{2.86M}{r_0(K_1 - 1)t^2}\left[\frac{3.33K_1^2 \log_{10} K_1 + 0.389(K_1^2 - 1)}{4.33K_1^2 + 2.33 + 1.18\,\dfrac{(K_1^2 - 1)t^3D}{\sqrt{r_0g^5}}}\right] \quad \ldots [4]$$

For the axial stress at c,

$$p_{h0} = p_{r0}\,(t/g)^2$$

And for the deflection of the ring portion of the flange,

$$y_1 = \frac{0.000{,}000{,}285\,M r_0}{(K_1 - 1)t^3}\left[\frac{K_1^2(\log_{10} K_1)^2}{K_1^2 - 1} + 0.0644(K_1^2 - 1) - \frac{0.0911(K_1^2 - 1) + 1.56\,K_1^2 \log_{10} K_1 + 6.67\,\dfrac{K_1^4}{K_1^2 - 1}\,(\log_{10} K_1)^2}{4.33\,K_1^2 + 2.33 + 1.18\,\dfrac{(K_1^2 - 1)t^3D}{\sqrt{r_0g^5}}}\right] \quad \ldots [5]$$

In the above, D represents a function of the dimensions h_1, g, and r_0, or "form function" for the hub, having the value

$$\frac{\cosh 2C + \cos 2C + 2}{\sinh 2C - \sin 2C} \quad \ldots [6]$$

where

$$C = \frac{\sqrt[4]{3(m^2 - 1)}}{\sqrt{mgr_0}}\,h_1 = 1.285\,\frac{h_1}{\sqrt{gr_0}} \quad \ldots [7]$$

This is a very awkward function, and the fact that it is an important part of an already complicated equation makes its use almost prohibitive in practical design. However, an approximation for D may be used that is much simpler, and at the same time is fairly accurate, as shown by the following table:

$C = 1.285\,\frac{h_1}{\sqrt{gr_0}}$	D (true value)	$D = \frac{1}{C^3} + 1$
0	∞	∞
1.0	1.971	2.000
1.4	1.187	1.186
1.8	1.036	1.053
2.2	1.019	1.019
2.6	1.016	1.008
3.0	1.014	1.004
∞	1.000	1.000

In most actual cases it will be found that for both high-hub and low-hub flanges, C lies between 1 and 3, in which range the approximation is especially close. For welding necks, and pipe with solid flanged ends, $C = \infty$ and the approximation again holds good. Only for large-diameter, low-hub flanges for low-pressure work, does C fall below 1, and the approximation then errs on the side of safety. Making this substitution in Equations [3], [4], and [5], the term

$$1.18\,\frac{(K_1^2 - 1)t^3D}{\sqrt{r_0g^5}} \quad \text{becomes} \quad (K_1^2 - 1)t^3\left(\frac{0.337r_0^2}{h_1^5} + \frac{1.18}{\sqrt{r_0g^5}}\right) \quad \ldots [8]$$

A comparison of Equations [1] and [3] with the simple flexure formula for beams brings out several interesting facts. In all cases the stress is directly proportional to the moment of the applied load, and inversely proportional to the square of the depth. If the length and breadth of a simple beam are increased in the same ratio, while the load and depth remain constant, the stress will also remain constant; similarly, if the inside and outside diameters of a flat ring are increased in the same ratio, and dimension b (Fig. 2) also increases in this ratio, while the depth and load remain constant, then the stress is unchanged. A 5-in. × 10-in. × 1-in. ring loaded with 50,000 lb. at the inner and outer edges has the same maximum stress as a 10-in. × 20-in. × 1-in. ring sustaining the same load. In this respect the new formulas correspond to those developed by both Tanner, and Crocker and Sanford.

When a hub is added to the flat ring, Equation [3] shows that the stress is computed by a formula identical with [2], except for the introduction of a new term which subtracts a greater or smaller amount from the result according as the hub height is large or small in relation to its diameter and thickness. If the hub height is zero, the denominator of this new term becomes infinite and it vanishes; if the hub height is very great, the term approaches a definite maximum value, indicating a stress that cannot be reduced by any possible extension of the hub. A similar relation holds between Equation [2] and Equation [5], which is the same as [2] with an additional term to give effect to the increased stiffness caused by the hub.

The equations given here represent conditions of stress and strain at particular points. General formulas for stress, deflection, slope, etc., at any point, are given in the Appendix. The theoretical stresses and deflection for a flange of the same proportions as that previously analyzed in Fig. 3, but with a hub added, are here plotted in Fig. 21. Note that, although the load is $3^1/_4$ times as great, the deflection is the same, and the maximum tangential stress is increased only 13 per cent.

TESTS ON HUBBED FLANGES

In order to ascertain definitely the great increase in strength and stiffness of hubbed flanges over plain rings, the American Spiral Pipe Works included low-hub and high-hub flanges in its 1924 and 1926 tests, previously alluded to. These tests also serve as an excellent check on formulas [3] and [5], and the results are submitted herewith, in Figs. 22 to 29, each figure representing the calculated deflection for a certain flange by means of a straight line, while the actual deflections as obtained in the testing machine are indicated by small dots and crosses. The agreement is fully as close as for the plain rings, and appears to be a sufficient guarantee that the formulas can be used to design flanges for conditions not yet covered by the standards.

RECOMMENDED PROPORTIONS OF HUBBED FLANGES

In conclusion, the following suggestions are offered for designing flanges with hubs, for cases not covered by existing standards:

1 Determine the bolt size and number as for plain rings.

2 Make the *minimum* bolt-circle diameter such that bolt holes are at least $2^1/_4$ diameters apart on centers, and determine the outside diameter as for plain rings.

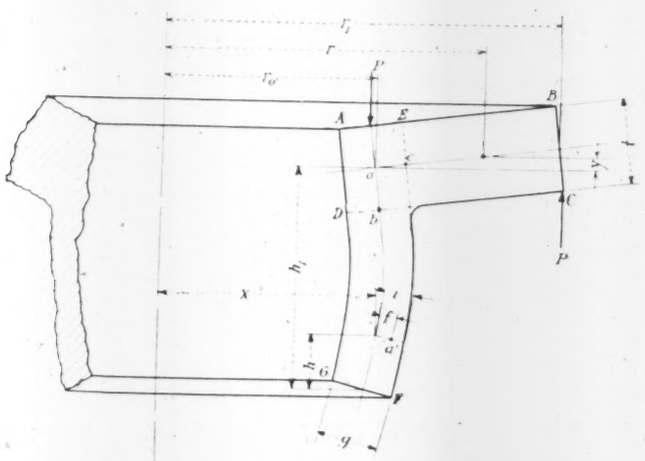


FIG. 20 DIAGRAM OF FLANGE DISTORTED UNDER TEST

3 Assuming that the flange is a cantilever beam supported at the center of pressure of the gasket contact, loaded at the bolt circle, and with a width equal to $\pi \times$ diameter at the center of pressure of gasket contact, compute the necessary thickness t by the simple flexure formula. Adjust the bolt-circle diameter, if necessary, to secure nut clearance around a hub of thickness g equal to $^1/_2 t$ to $^3/_4 t$, and modify t correspondingly.

4 Assume a hub height $h_1 = 1.5\sqrt{r_0 g}$; examination of the equation for p_t shows that any increase in h_1 above this amount adds practically nothing to the strength of the flange. It will also be noted that this proportion results in making the hub height h_1 from 2 to 3 times the flange thickness t. Then substitute all the assumed dimensions in Equation [3], and make a trial solution for p_{t0}. If this is too great or too small, adjust t, g, or h_1 until the proper stress is secured.

NOTE: In designing a flanged fitting or welding neck, where the hub not only reinforces the ring but withstands internal pressure as well, the total hoop stress will be the sum of the stress p_{t0} as above, and the stress due to fluid pressure as computed by the modified Barlow formula.[8]

[8] See A.S.M.E. Boiler Code, sec. P-23.

EXAMPLE: Design a hubbed flange for 12-in. pipe and 600 lb. per sq. in. working steam pressure. Assume the following proportions and values:

Gasket pressure = total steam pressure
Working stress in bolts = 14,000 lb. per sq. in.
Working stress in flange, maximum = 14,000 lb. per sq. in.
Diameter of raised face = 15 in.
Bolt size = $1^1/_4$ in., with 8 Am. std. threads per inch
Bolt-hole diam. = $1^3/_8$ in.

The total steam pressure acting on a 15-in. circle is 106,200 lb., so that the total bolt load is 212,400 lb. The number of bolts theoretically required is 16.4, which may be changed to 20 so as to have a multiple of four.

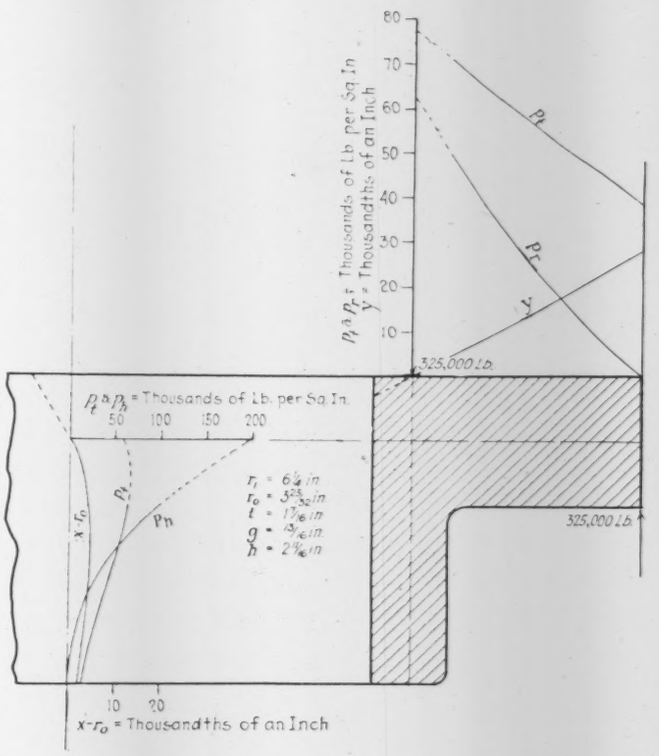

p_t & p_r = Thousands of Lb. per Sq. In.
y = Thousandths of an Inch
325,000 Lb.
p_t & p_r = Thousands of Lb. per Sq. In.
r_i = 6¼ in
r_0 = 3²⁵/₃₂ in
t = 17/16 in
g = 15/16 in
h = 2¹⁵/₁₆ in
$x-r_0$ = Thousandths of an Inch

FIG. 21 PLOTTINGS OF THEORETICAL STRESSES AND DEFLECTION OF A HUBBED FLANGE

Twenty $1^3/_8$-in. holes, spaced $2^1/_4$ diameters, or 3.09 in., on centers, require a 19.7-in. bolt circle; assume a minimum bolt-circle diameter of $19^{11}/_{16}$ in. The inside diameter of the flange will be $12^3/_4$ in., to accommodate the outside diameter of 12-in. pipe, and the outside diameter of the flange may be made 22 in., which will give full seating for U. S. std. nuts with about $^1/_{32}$ in. to spare.

The center of pressure of gasket contact comes at the middle of the raised face, i.e., on a $13^7/_8$-in. circle. The moment of loading is therefore $212{,}400 \left(\frac{19^{11}/_{16} - 13^7/_8}{2}\right)$ or 616,000 lb-in.

The width of the so-called "cantilever" is $\pi \times 13^7/_8$, the bending moment is 616,000 lb-in., and the maximum working stress is 14,000 lb. per sq. in.; hence, by the simple flexure formula, the tentative flange thickness t is 2.46 in. We may now assume a hub thickness g which bears some arbitrary relation to the flange thickness—say $^5/_8 t$, making $g = 1^9/_{16}$ in. This gives a hub diameter of $15^7/_8$ in., and the distance from outside of hub to bolt circle is well in excess of half the bolt spacing, so that there is ample nut clearance and space for a fillet at the base of the hub.

Assuming that $h_1 = 1.5\sqrt{g r_0}$ or $4^3/_4$ in., and substituting the tentative values for loading moment, t, g, h_1, etc., in formula [3], it is found that the maximum tangential stress $p_{t0} = 9940$ lb. per sq. in. This indicates that the design is too strong for the given conditions. By rounding off t to $2^1/_2$ in. and decreasing h_1 to $2^1/_2$ in., the stress is increased to 13,130 lb. per sq. in., and a

design is obtained which approximates the American Standard low-hub flange for 600 lb. working steam pressure.

Obviously it would be better in the present instance to make the flange agree in all respects with the standard, inasmuch as a standard does exist for the size of pipe and steam pressure that were assumed in the statement of the problem. However, the solution is interesting as an illustration of the possibilities of the method.

Numerous other uses can be found for the stress formula, e.g., the calculation of the hub thickness and height that will add the maximum possible strength to a ring of given diameter, bore, thickness, and bolting; the determination of the proper proportions for hub height and ring thickness to secure maximum strength for a given total weight; comparison of strength of hubbed flanges and plain rings of equal weight, etc. Space permits nothing more than a brief mention of these various applications, in a paper of this scope; however, they are offered to the manufacturer and user of flanges and flanged fittings, with confidence that they will be found useful in many cases that are outside the field covered by accepted or tentative standards for dimensions and materials.

In conclusion, the authors wish to call especial attention to the tests carried out by the American Spiral Pipe Works. It is believed that they contribute a body of data that is not only unique in this field, but is also of broader scope than has hitherto been contemplated by manufacturers and users of flanges. It would be possible to make them the basis for an empirical method of design, quite independently of the theory that has been developed in this paper. Taken with the theoretical formulas, it is to be hoped that they form a solid foundation upon which future design can be built, corroborating the standard proportions that have already been adopted or tentatively proposed, and furnishing the necessary data for the vast number of special sizes for which no standards are being developed.

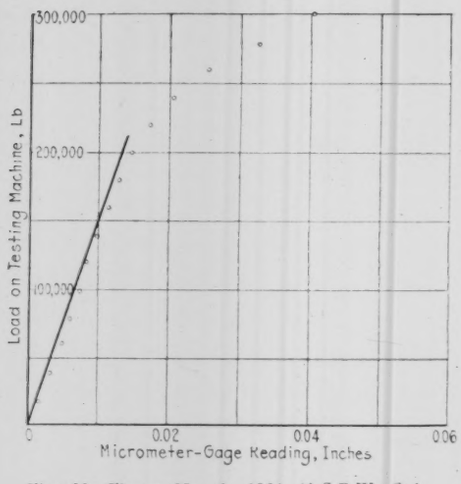


Fig. 22—Flange No. 1, 1924 (A.S.P.W. Std. 6-in. 250-lb. high-hub flange), $6^{9}/_{8}$ in. I. D., $12^{1}/_{2}$ in. O. D., $1^{1}/_{4}$ in. flange thickness, $8^{1}/_{4}$ in. hub diameter, $3^{1}/_{4}$ in. overall height, standard drilling for bolt holes.

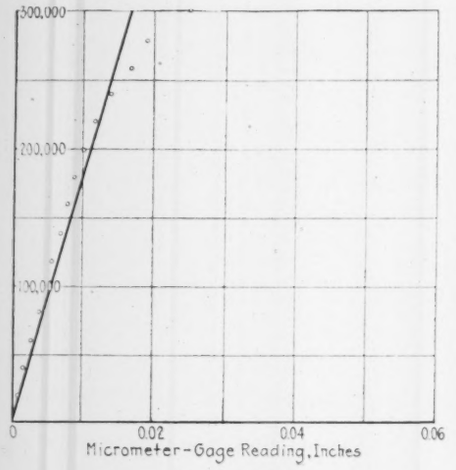


Fig. 23—Flange No. 2, 1924 (Am. Std. 6-in. diameter 250-lb. high-hub flange), $6^{9}/_{8}$ in. I. D., $12^{1}/_{2}$ in. O. D., $1^{7}/_{16}$ in. flange thickness, $8^{1}/_{4}$ in., hub $3^{7}/_{16}$ in. overall height, standard drilling for bolt holes.

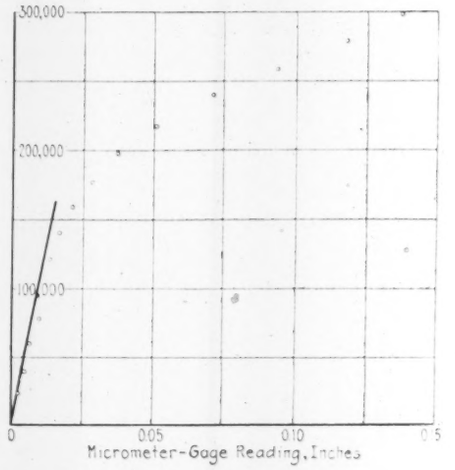


Fig. 24—Flange No. 3, 1924 (A.S.P.W. Std. 6-in. 250-lb. low-hub flange bored to $6^{9}/_{8}$ in.[1]), $6^{9}/_{8}$ in. I. D., $12^{1}/_{2}$ in. O. D., $1^{1}/_{4}$ in. flange thickness, $8^{1}/_{4}$ in. hub diameter, 2 in. overall height, standard drilling for bolt holes.

[1] Hub about 50 per cent higher than minimum A.E.S.C. Specifications.

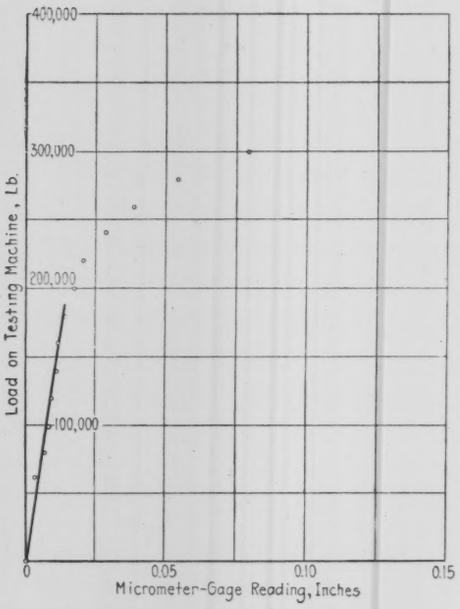


Fig. 25—Flange No. 3-A, 1924 (Am. Std. 6-in. 250-lb. low-hub flange bored to $6^{9}/_{8}$ in.), $6^{9}/_{8}$ in. I. D., $12^{1}/_{2}$ in. O. D., $1^{7}/_{16}$ in. flange thickness, $8^{1}/_{4}$ in. hub diameter, $2\ ^{3}/_{16}$ in. overall height, standard drilling for bolt holes.

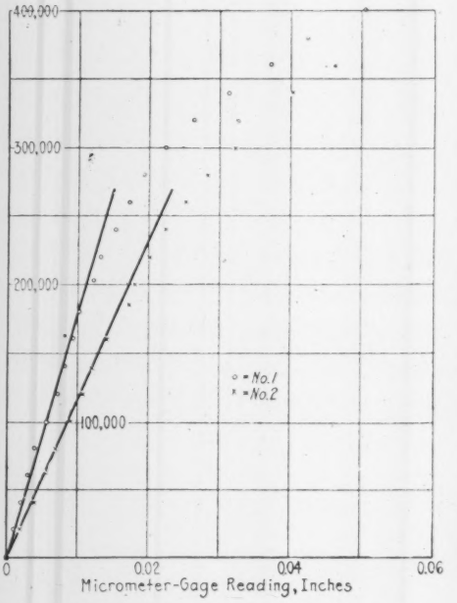


Fig. 26—Flanges Nos. 1 and 2, 1926 (A.S.P.W. Std. 10-in. 250-lb. low-hub flange bored to $10^{3}/_{4}$ in.[1]), $10^{3}/_{4}$ in. I. D., $17^{1}/_{2}$ in. O. D., $1^{1}/_{2}$ in. flange thickness, $12^{5}/_{8}$ in. hub diameter, $2^{3}/_{8}$ in. overall height, standard drilling for bolt holes.

[1] Hub about 50 per cent higher than minimum A.E.S.C. Specifications.

NOTE: Two distinct plots are due to difference in length of outer gage bar for flange Nos. 1 and 2.

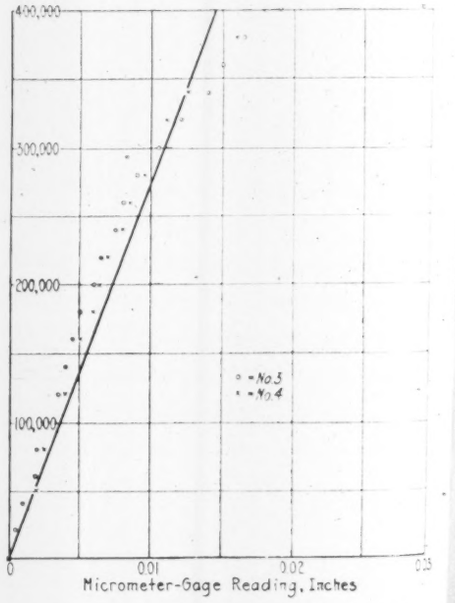


Fig. 27—Flanges Nos. 3 and 4, 1926 (A.S.P.W. Std. 10-in. 250-lb. high-hub flange), $10^{3}/_{4}$ in. I. D., $17^{1}/_{2}$ in. O. D., $1^{1}/_{2}$ in. flange thickness, $12^{5}/_{8}$ in. hub diameter, $3^{3}/_{4}$ in. overall height, standard drilling for bolt holes.

FIGS. 22–27 TESTS OF HUBBED FLANGES, AMERICAN SPIRAL PIPE WORKS, 1924–1926

Appendix

I—FORMULAS FOR PLAIN RINGS

Let the constants A' and A'' be determined as follows:

$$A' = \frac{K_1^2}{K_1^2 - 1} \log_e K_1 + \frac{1 + 3m}{2(1 + m)}$$

$$A'' = \frac{2(1 + m)K_1^2}{(m - 1)(K_1^2 - 1)} \log_e K_1$$

then

$$y = \frac{3(m^2 - 1) P r_0^2}{2\pi m^2 E t^3} [A'(K^2 - 1) - (\log_e K)(K^2 - A'')]$$

$$\frac{dy}{dr} = \frac{3(m^2 - 1) P r_0}{2\pi m^2 E t^3} \left[\left(2A' - 2\log_e (K - 1) K + \frac{A''}{K}\right)\right]$$

$$\frac{d^2y}{dr^2} = \frac{3(m^2 - 1) P}{2\pi m^2 E t^3} \left[2A' - 3 - 2\log_e K - \frac{A''}{K^2}\right]$$

$$\frac{d^3y}{dr^3} = \frac{3(m^2 - 1) P}{2\pi m^2 E t^3 r_0} \left[\frac{2A''}{K^3} - \frac{2}{K}\right]$$

$$p_t = -\frac{3P}{4\pi m t^2} \left[2(m + 1)(A' - \log_e K) + (m - 1)\frac{A''}{K^2} - (m + 3)\right] \text{ when } l = \frac{t}{2}$$

$$= p_{t\,\max} \text{ when } K = 1 \text{ and } l = -\frac{t}{2}$$

$$p_r = -\frac{3P}{4\pi m t^2} \left[2(m + 1)(A' - \log_e K) - (m - 1)\frac{A''}{K^2} - (3m + 1)\right] \text{ when } l = \frac{t}{2}$$

$$= p_{r\,\max} \text{ when } K = K_1 \sqrt{\frac{2\log_e K_1}{K_1^2 - 1}} \text{ and } l = -\frac{t}{2}$$

$$s = \frac{3P}{4\pi t r_0 K} \text{ when } l = 0$$

$$= s_{\max} \text{ when } K = 1.$$

II—FORMULAS FOR HUBBED FLANGES

Let the constants B' and B'' be determined as follows:

$$B' = (m - 1) r_0 n + \frac{3P(m^2 - 1)}{2\pi m^2 E t^3} \times \left[\frac{2(m - 1) r_0^2 \log_e r_0 + 2(m + 1) r_1^2 \log_e r_1 + (m - 1)(r_1^2 - r_0^2)}{(m - 1) r_0^2 + (m + 1) r_1^2}\right] \ldots [a]$$

$$B'' = r_0 n + \frac{3P(m^2 - 1) r_0^2}{2\pi m^2 E t^3} (2\log_e r_0 - 1) - B' r_0^2 \ldots [b]$$

where $n = \frac{dy}{dr}$ when $r = r_0$; then

$$y = \frac{3P(m^2 - 1)}{2\pi m^2 E t^3} (r^2 - r_0^2 + r_0^2 \log_e r_0 - r^2 \log_e r) + \frac{B'}{2}(r^2 - r_0^2) + B'' \log_e \frac{r}{r_0}$$

$$\frac{dy}{dr} = \frac{3P(m^2 - 1)}{\pi m^2 E t^3} \left(\frac{r}{2} - r\log_e r\right) + B' r + \frac{B''}{r}$$

$$\frac{d^2y}{dr^2} = -\frac{3P(m^2 - 1)}{\pi m^2 E t^3} \left(\frac{1}{2} + \log_e r\right) + B' - \frac{B''}{r^2}$$

$$\frac{d^3y}{dr^3} = -\frac{3P(m^2 - 1)}{\pi m^2 E t^3 r} + \frac{2B''}{r^3}$$

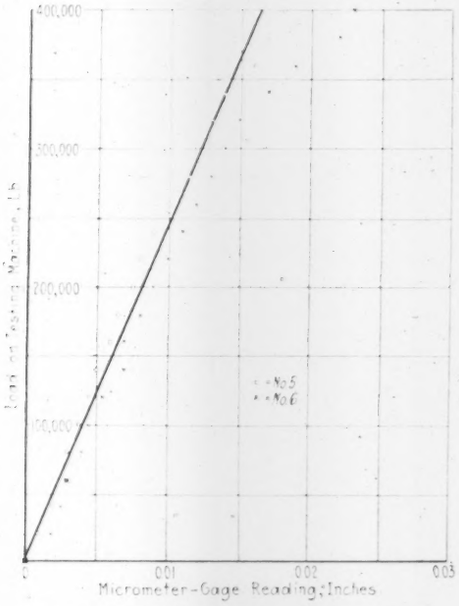


Fig. 28—Flanges Nos. 5 and 6, 1926 (Am. Std. 10-in. 250-lb. low-hub flange bored to $10^3/_4$ in.), $10^3/_4$ in. I. D., $17^1/_2$ in. O. D., $1^7/_8$ in. flange thickness, $12^5/_8$ in. hub diameter, $2^3/_4$ in. overall height, standard drilling for bolt holes.

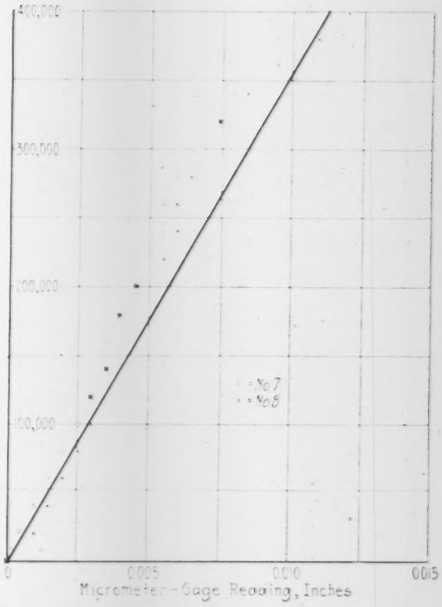


Fig. 29—Flanges Nos. 7 and 8, 1926 (Am. Std. 10 in. 250-lb. high-hub flange), $10^3/_4$ in. I. D., $17^1/_2$ in. O. D., $1^7/_8$ in. flange thickness, $12^5/_8$ in. hub diameter, $4^1/_8$ in. overall height, standard drilling for bolt holes.

FIGS. 28 AND 29 TESTS OF HUBBED FLANGES, AMERICAN SPIRAL PIPE WORKS, 1924–1926

$$p_t = \frac{3P}{4\pi m t^2} \left[(m - 1) - 2(m + 1)\log_e r\right] + \frac{mEt}{2(m^2 - 1)} \left[(m + 1)B' + (m - 1)\frac{B''}{r^2}\right] \text{ when } l = -\frac{t}{2}$$

$$p_r = \frac{3P}{4\pi m t^2} \left[(m - 1) + 2(m + 1)\log_e r\right] + \frac{mEt}{2(m^2 - 1)} \left[(m + 1)B' - (m - 1)\frac{B''}{r^2}\right] \text{ when } l = -\frac{t}{2} \ldots [c]$$

$$\mathbf{M}_0 = \frac{p_{r0} t^2}{6} \ldots [d]$$

Also,

$$\mathbf{M}_0 = -\frac{m^2 E g^3}{12(m^2 - 1)} \frac{d^2x}{dh^2} \text{ when } h = h_1.$$

Let the constants C, C', C'', C''', be determined as follows:

$$C = \sqrt[4]{\frac{3(m^2 - 1)}{m^2 g^2 r_0^2}}$$ h_1, h_1 being taken as +

$$C' = -\frac{\sqrt{12(m^2 - 1)}}{mEg^2} r_0 \mathbf{M}_0 \frac{\cos C \cosh C}{\sinh 2C - \sin 2C}$$

$$C'' = \frac{\sqrt{12(m^2 - 1)}}{mEg^2} r_0 \mathbf{M}_0 \frac{\sin C \cosh C + e^c \cos C}{\sinh 2C - \sin 2C}$$

$$C''' = \frac{\sqrt{12(m^2 - 1)}}{mEg^2} r_0 \mathbf{M}_0 \frac{\sin C \cosh C - e^{-c} \cos C}{\sinh 2C - \sin 2C}$$

then

$$x - r_0 = e^{\frac{Ch}{h_1}} \left(C' \sin\frac{Ch}{h_1} + C'' \cos\frac{Ch}{h_1}\right) + e^{-\frac{Ch}{h_1}} \left(C' \sin\frac{Ch}{h_1} + C'' \cos\frac{Ch}{h_1}\right)$$

$$\frac{dx}{dh} = \frac{C}{h_1} \left\{e^{\frac{Ch}{h_1}} \left[C' \left(\sin\frac{Ch}{h_1} + \cos\frac{Ch}{h_1}\right) - C''' \left(\sin\frac{Ch}{h_1} - \cos\frac{Ch}{h_1}\right)\right]\right.$$

$$- e^{-\frac{Ch}{h_1}}\left[C'\left(\sin\frac{Ch}{h_1} - \cos\frac{Ch}{h_1}\right) + C''\left(\sin\frac{Ch}{h_1} + \cos\frac{Ch}{h_1}\right)\right]\Bigg\}$$

$$\frac{d^2x}{dh^2} = \frac{2C^2}{h_1^2}\left[e^{\frac{Ch}{h_1}}\left(C'\cos\frac{Ch}{h_1} - C''\sin\frac{Ch}{h_1}\right)\right.$$

$$\left. - e^{-\frac{Ch}{h_1}}\left(C'\cos\frac{Ch}{h_1} - C''\sin\frac{Ch}{h_1}\right)\right]$$

$$\frac{d^3x}{dh^3} = \frac{2C^3}{h_1^3}\left\{e^{\frac{Ch}{h_1}}\left[C'\left(\cos\frac{Ch}{h_1} - \sin\frac{Ch}{h_1}\right) - C''\left(\cos\frac{Ch}{h_1} + \sin\frac{Ch}{h_1}\right)\right]\right.$$

$$\left. + e^{-\frac{Ch}{h_1}}\left[C'\left(\cos\frac{Ch}{h_1} + \sin\frac{Ch}{h_1}\right) + C''\left(\cos\frac{Ch}{h_1} - \sin\frac{Ch}{h_1}\right)\right]\right\},$$

$$p_h = \frac{mE}{m^2-1}\left[-mf\frac{d^2x}{dh^2} + \frac{x-r_0}{r_0}\right]$$

$$= -\frac{m^2Eg}{2(m^2-1)}\frac{d^2x}{dh^2} \text{ when } f = i,$$

$$p_t' = \frac{mE}{m^2-1}\left[\frac{m}{r_0}(x-r_0) - f\frac{d^2x}{dh^2}\right].$$

$$= E\frac{x-r_0}{r_0} - \frac{mEg}{2(m^2-1)}\frac{d^2x}{dh^2} \text{ when } f = i;$$

$$u = \frac{6(m^2-1)^{3/4}r_0^{1/2}}{3^{1/4}m^{3/2}Eg^{5/2}}\,\mathbf{M}_0\,\frac{\cosh 2C + \cos 2C + 2}{\sinh 2C - \sin 2C} \qquad [e]$$

Substituting Equations [a] and [b] in [c], and substituting p_{r0} for p_r and r_0 for r, an equation is obtained with p_{r0} and u as unknown quantities. Substituting this in Equation [d] gives one relation between $\mathbf{M}_0$ and n, which, solved simultaneously with [e], gives

$$\mathbf{M}_0\left[1 + \frac{t^3(r_1^2-r_0^2)(m^2-1)^{3/4}(\cosh 2C + \cos 2C + 2)}{2\times 3^{1/4}m^{1/2}r_0^{1/2}g^{5/2}[(m+1)r_1^2 + (m-1)r_0^2](\sinh 2C - \sin 2C)}\right]$$

$$= \frac{P}{2\pi}\left[\frac{(m+1)r_1^2\log_e\frac{r_1}{r_0} + \frac{m-1}{2}(r_1^2-r_0^2)}{(m+1)r_1^2 + (m-1)r_0^2}\right]$$

Some Uses of the High-Speed Multi-Cylinder Indicator

Particulars of a Simple, Rugged Device for Use on High-Speed Internal-Combustion Engines and Air Compressors—Diagrams Obtained—Multi-Unit Development for Obtaining Diagrams Simultaneously from All Cylinders of Engine or Compressor

By H. M. JACKLIN,[1] COLUMBUS, OHIO.

THERE being a very apparent need for an indicator that was simple, easily used, and which gave records that were permanent and of immediate use in the development of high-speed reciprocating machinery, the author with Mr. C. P. Roberts[2] started

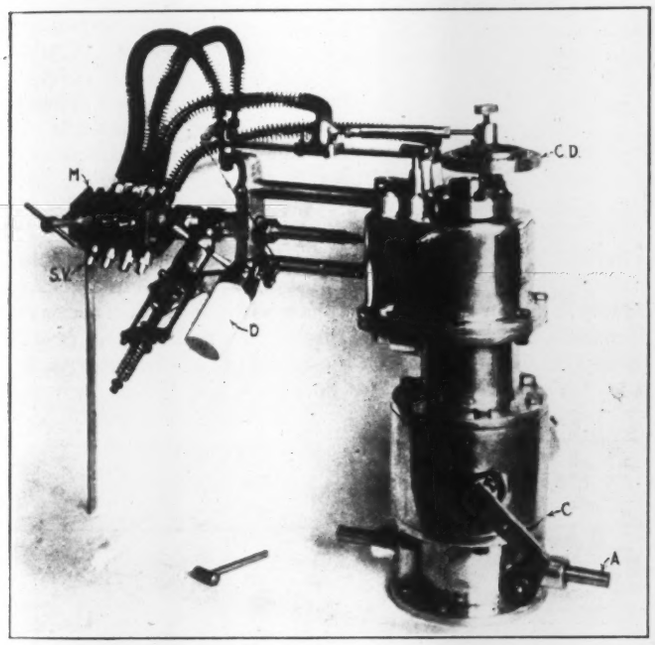


FIG. 1 THE AUTHOR'S SINGLE-UNIT HIGH-SPEED MULTI-CYLINDER INDICATOR

on the development of such an instrument some few years ago. It was felt that, with the proper driving mechanism, a simple and reliable low-speed indicator such as the Crosby could be made to obtain such results. Accordingly the mechanism as shown in Fig. 1 was developed. This figure shows the complete assembly of the instrument with tubes for connecting to the several cylinders of a four-cylinder engine. There is a small poppet valve arranged between the cylinder of the indicator and the cylinder of the engine. This valve is opened for a small interval of time in each cycle of the engine, so that a very small amount of gas or fluid may pass into or from the indicator cylinder in order that the pressure may become balanced. The shaft *A* of the instrument is ordinarily driven at the same speed as the crankshaft of the engine being tested through a suitable positive drive. Eight hundred cycles of a four-stroke engine are required to obtain a complete diagram, while only four hundred cycles are required on a two-stroke engine or an air compressor. In other words, the resulting diagram shows a composite of two hundred cycles for each stroke of the engine. The pressures are built up and let down in small increments while the card is moved back and forth at a very slow speed. Thus, except where the pressures vary greatly from cycle to cycle—as will later become apparent—there is no inertia entering into the motion of the pencil mechanism of the low-speed indicator. Nor can any inertia be present in the operation of the drum of the low-speed indicator, since it is moved at a very slow speed.

The motion of this drum *D* is controlled by a string passing over suitably located pulleys to the small crosshead at the top of the device. This crosshead is in turn controlled by the connecting-rod–crank train shown, said crank rotating once for eight hundred revolutions of a four-stroke engine and once for four hundred revolutions of a two-stroke engine. Since this connecting rod is adjustable in length, the ratio of its length to the throw of its crank can be made exactly the same as that of the machine under test. Further, the crosshead guide may be adjusted to simulate the offset in the cylinders of the engine, should the engine be of that type.

The cylinder of the indicator is ordinarily filled with a heavy lubricant to reduce the clearance and to provide an effective oil seal. The reduced clearance means that only an almost infinitesimal amount of working fluid need enter or leave the indicator in order to balance the pressure, while an effective oil seal is necessary since the pencil must be held stationary between the instants that the valve is open.

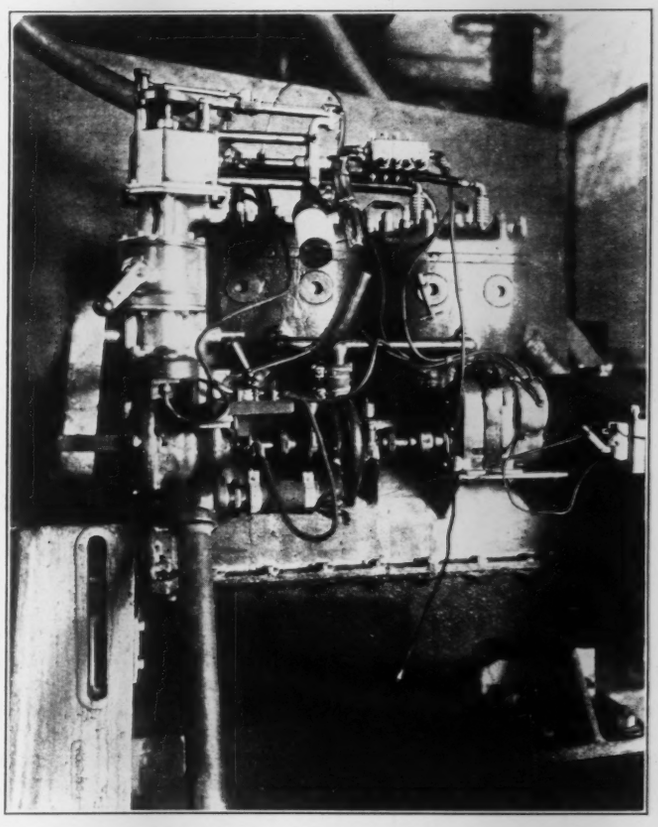


FIG. 2 INDICATOR MOUNTED ON A FOUR-CYLINDER WAUKESHA ENGINE AT PURDUE UNIVERSITY

The graduated disk *CD*, on which the small crank is mounted, and the collector manifold *M* are provided so that the device may be used on a multi-cylinder engine. The disk enables the operator to move the small crank so that it simulates any crankpin of the engine; while the collector manifold enables him to connect the

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Be sure to bring this copy with you to the Spring Meeting

valve chamber to the corresponding cylinder by means of the small manually operated selection valves *SV*.

The hand crank *C* may be used to operate the crank disk in either direction and at whatever speed is desired. In operation, this hand crank is pulled or pushed axially so that a slow-speed jaw clutch is engaged inside the mechanism. This makes for uniformity in results, as the whole mechanism is then driven from the machine under test.

It is apparent that the whole device is rather rugged. In fact, it was used to obtain diagrams on a moving automobile at the Semi-

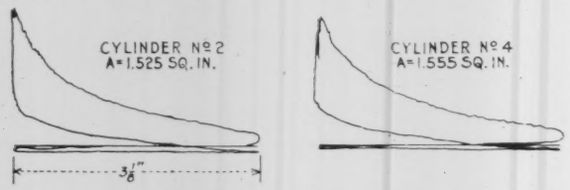


FIG. 3 DIAGRAMS OBTAINED FROM WAUKESHA ENGINE AT PURDUE UNIVERSITY

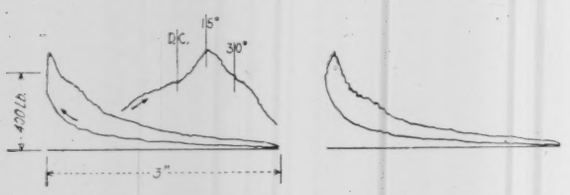


FIG. 4 DIAGRAMS OBTAINED FROM A DIESEL ENGINE RUNNING AT 600 R.P.M. AT PLANT IN CLINTON, IOWA

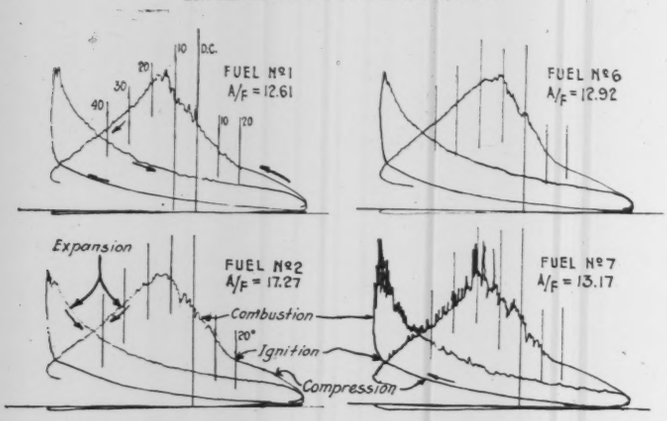


FIG. 5 DIAGRAMS SHOWING THE BEHAVIOR OF SEVERAL DIFFERENT GASOLINES IN A SMALL ENGINE RUNNING AT 1090 R.P.M.

Annual Meeting of the Society of Automotive Engineers in June, 1926. Diagrams have been taken from all four cylinders of an automobile on the road at speeds up to 25 miles per hour. It was not then apparent that any limit had been reached.

DIAGRAMS OBTAINED WITH DEVICE

Since the device was described in some detail in the *Journal of the S.A.E.* for June, 1926, it is not deemed necessary to do so in this paper. One installation, that on a Waukesha engine at Purdue University and shown in Fig. 2, is a very excellent one wherein the collector manifold is well located with reference to the cylinders of the engine. Fig. 3 shows some very excellent diagrams obtained from this engine running at 1200 r.p.m. One of these diagrams is from cylinder No. 2, while the other is from cylinder No. 4, of the Waukesha engine. The engine was operating at less than full load, as will be apparent from the size of the lower loops in the two diagrams; it should also be noted that the combustion line is rather clear in both diagrams, probably showing some of the effects of turbulence within the cylinders.

Fig. 4 shows two diagrams obtained from a two-cycle Diesel engine at the plant of the Climax Engineering Co., Clinton, Iowa. These were furnished by Mr. R. F. Gagg, assistant Diesel engineer. They have been carefully traced from the original diagrams and show the capabilities of the device on such an engine running at 600 r. p. m. The offset diagram at the left certainly conveys information that is more accurate than can be obtained where the drum of the indicator is operated manually, since exact crank-angle positions may be shown. Thus, combustion processes may be studied with rather great accuracy. Changes in the injection time show very clearly, while the variations between cylinders become apparent.

It was hoped to have a photograph and some sample diagrams of an installation on a high-speed air compressor. These were not obtainable at this time. However, diagrams were successfully obtained at speeds corresponding to 2750 to 2960 r.p.m. on an automobile engine. In this installation a rather short tube was used to connect to the high-pressure cylinder, while longer connecting tubes were used to the low-pressure cylinders. These tubes are of commercial copper tubing $^{5}/_{16}$ in. in outside diameter and have proved very satisfactory in that installation.

Fig. 5 comprises part of a series of diagrams showing the action of several different fuels in a 2 $^{1}/_{2}$-in. × 5-in. Delco-Light engine running at 1090 r.p.m. The ignition was set to occur 20 deg. in advance, and the volumetric compression ratio was 4.47. It will be noted that fuel No. 7 shows spikes on its diagrams, indicating detonation. It is not probable that all the pressure is shown for the peaks of these spikes, due to the inertia of the pencil motion. With few exceptions these spikes were made with the sampling valve open and do not add area to the diagrams. Fuel No. 6 seems to be particularly smooth-burning, since its combustion lines are very smooth. These diagrams bring out one of the features of the device. The conventional power diagram is readily recognized. By simply moving the small crank at the top of the instrument so that it is 75 deg. or 90 deg. out of phase with the crank of the engine, it is possible to obtain the offset diagrams shown. The vertical line marked D.C. is the position of the card when the engine is on dead center, while the other shorter lines show the 10-deg. increments

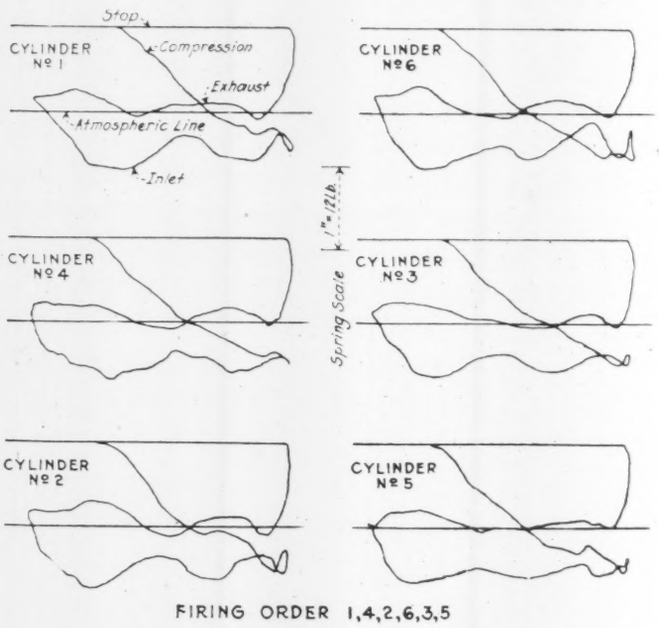


FIG. 6 LOWER-LOOP DIAGRAMS OBTAINED FROM A SIX-CYLINDER ENGINE RUNNING AT 1200 R.P.M.

before or after dead center. This offset diagram makes it possible to minutely examine any particular part of the diagram, since the abscissas of that part can be increased. In this case the ignition line is spread over a horizontal distance of about one inch. The small variations shown in this line are due to variations from cycle to cycle, which disappear almost entirely when two or more spark plugs are used or when considerable turbulence is present in the combustion chamber.

Fig. 6 shows some interesting lower-loop diagrams taken from a six-cylinder engine using a single carburetor and what is considered an excellent type of manifold. These diagrams were procured in less than ten minutes, the engine running at 1200 r.p.m. at about one-third load. Assuming that the charge is heated equally in all

cylinders, the volume in the cylinder measured on the atmospheric line should be a measure of the relative volumetric efficiencies of the several cylinders. On this basis the volumetric efficiency of the various cylinders was as follows: No. 1, 60.6; No. 2, 60.6; No. 3, 65.7; No. 4, 59.6; No. 5, 58.6; and No. 6, 56.5 per cent. The maximum variation was 9.2 points. It is apparent that No. 3 got the larger charge. Power diagrams showed that this cylinder was delivering the greatest indicated horsepower. These lower-loop diagrams show the volumetric distribution very well, while offset power diagrams would give a pretty accurate measure of the quality distribution, since lean mixtures show a flatter combustion line than others. All six of these diagrams show peculiarities that are not apparent in the lower loops of the two diagrams in Fig. 3, which were taken on the four-cylinder engine. It is manifest that there must be some disturbing factor in the six-cylinder engine.

CRANK-ANGLE DIAGRAMS

Fig. 7 shows a crank-angle pressure diagram as worked out form the lower-loop diagrams in Fig. 6, with the several diagrams arranged in their proper order. This composite diagram shows that the pressures *within* the cylinders must be affected by pulsations in the manifolds, and that these pulsations have a frequency about double that of the cycles, since nodes occur in the cylinders at about every sixty degrees. It appears, further, that these pulsations act to rob cylinder No. 6 and to load cylinder No. 3, with the resulting effects already mentioned regarding these cylinders. It is regretted that time did not permit carrying these particular tests further at other speeds and loads, as one is left wondering (1) whether these conditions obtain throughout the entire range of both speed and power, (2) whether they become more aggravating, or (3) whether they smooth out and disappear entirely.

The question of tube lengths will probably be raised as relating to the accuracy of the diagrams obtained where the pressures are rather high. There is of course some little lag in pressure effect in a tube of any considerable length. With 1/4-in. outside-diameter copper tubing considerable lag was observed in lengths above 18 in. on the Delco-Light engine running at 1200 r.p.m. The 18-in. length of tube gave a diagram which was almost an exact duplicate of one obtained with a tube 13 in. in length, which was then the minimum length that it was possible to use. Since then the assembly has been changed so that we now use a tube about three inches long which has not shown any material variation from the longer tube. On a multi-cylinder engine where all tubes are made of

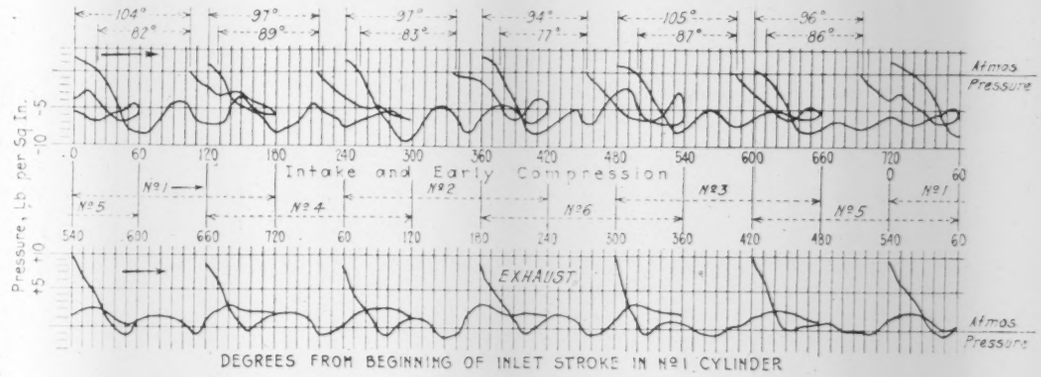


FIG. 7 COMPOSITE OF VARIATIONS IN PRESSURE WITHIN THE CYLINDER OF A SIX-CYLINDER ENGINE RUNNING AT 1200 R.P.M. AND ABOUT ONE-QUARTER LOAD DURING EXHAUST AND INLET STROKES

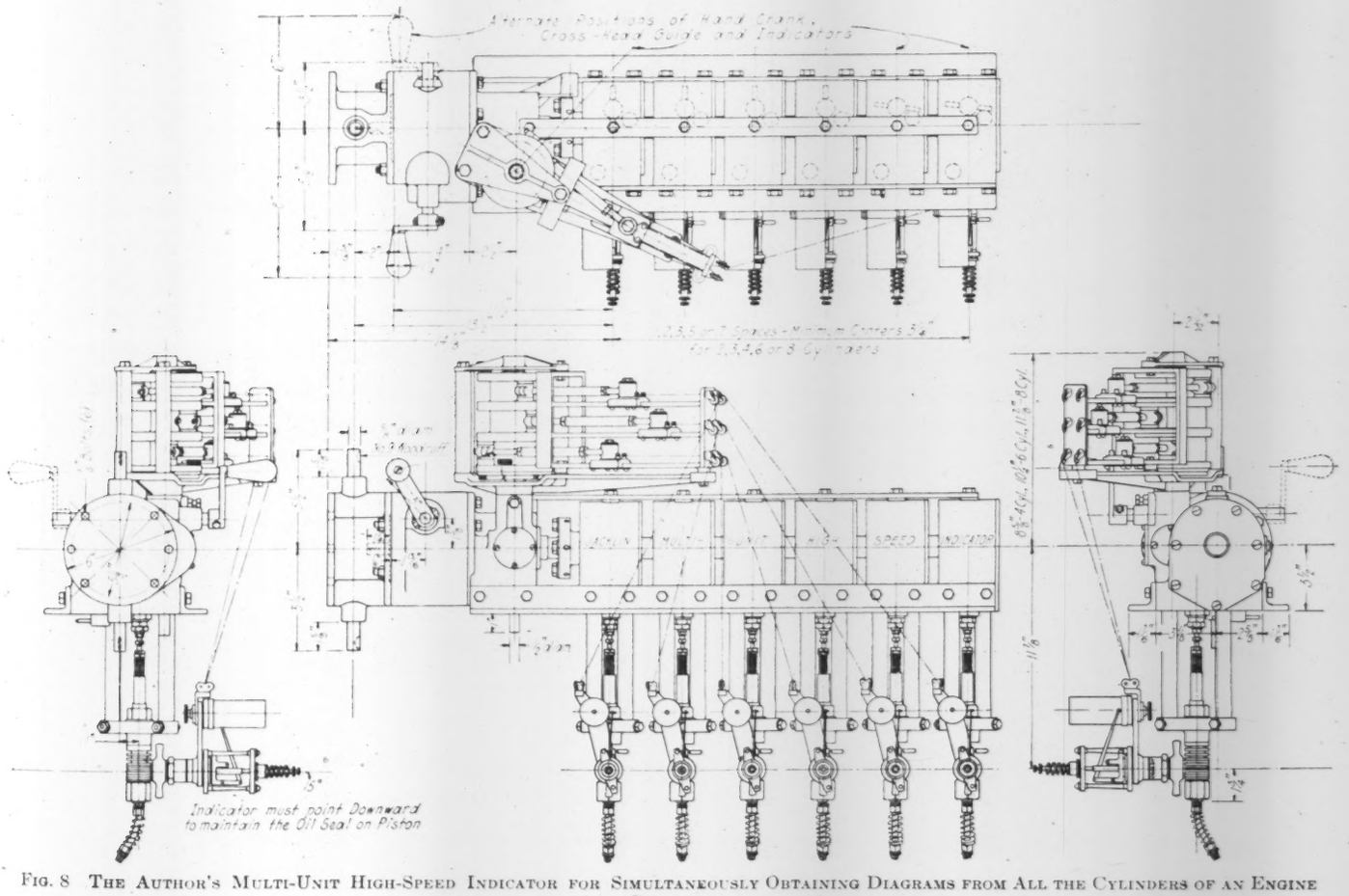


FIG. 8 THE AUTHOR'S MULTI-UNIT HIGH-SPEED INDICATOR FOR SIMULTANEOUSLY OBTAINING DIAGRAMS FROM ALL THE CYLINDERS OF AN ENGINE OR COMPRESSOR

exactly the same length and with long, smooth bends, direct comparisons can be made between the various cylinders with the single-unit instrument shown in Fig. 1, although the tubes may be long enough to dampen out some of the high pressures. The diagrams are of course taken from but one cylinder at a time, and some care must be exercised by the operator to maintain the conditions constant during the tests.

Multi-Unit High-Speed Indicator for Simultaneous Use on All Cylinders

Fig. 8 shows a development of the device that presents several advantages. It is a multi-unit high-speed indicator wherein diagrams may be simultaneously obtained from all the cylinders of an engine or compressor. Its design is such that it can be made to go on almost any type of engine. A further advantage is that the connecting tubes can be made very short indeed, if not eliminated entirely. It will be apparent that it is simply an expansion of the instrument shown in Fig. 1, more sampling valves and slow-speed indicators being added. A crankshaft is also necessary so that the several drums may be in phase with the crankpins of the several cylinders of the engine. The control is exactly the same as in the single-unit instrument where diagrams could only be obtained successively from the several cylinders. Necessarily, such an instrument as shown in Fig. 8 must be "tailor made" for a particular engine in order to have the connecting tubes as short as possible, and in order to have the sampling valves and indicator drums properly synchronized with the cycles of the several cylinders. On this account the valve units are made up separately and assembled between the machined angles as shown, the minimum center distance between valves being $3^1/_4$ in. In some special cases it may be necessary to do away with these angles and support the sampling-valve–indicator assembly on its particular engine cylinder, as in the case of several high-speed oil engines.

Fig. 8 shows that each crosshead controls two indicator drums, as would be the case in a four-stroke engine. Most two-stroke engines would require a crosshead for each cylinder.

The single-unit device has been used to study distribution of mixture, cylinder performance, multiple ignition, and fuels in internal-combustion engines; to study a high-speed air compressor; and one is now in use on a Diesel engine. It is entirely practicable to use the complete device to do anything that one would expect to do with a low-speed indicator on a low-speed engine. Actually, it is oftentimes easier to use the instrument on a high-speed engine than to use the low-speed indicator alone on an engine running under 300 r.p.m. In any case, the resulting diagrams show average conditions more nearly than is possible by averaging many single diagrams, since a composite diagram of many cycles will show average conditions with greater accuracy.

A Uniform Apprenticeship Certificate for Co-operating Groups of Employers

By WM. S. CONANT,[1] WASHINGTON, D. C.

THE revival of apprenticeship in American industry may be said to have passed through two phases and be now entering a third.

First, that of skepticism. Every one associated the idea of apprenticeship with the obsolete conditions of the family factory, where pride in the craftsmanship of the product existed and where a paternal interest was felt in the young men who came to learn those painstaking methods of fabrication practiced by the preceding generation of workmen. Recognizing that those conditions had departed forever, it was assumed that they had carried with them the trade apprentice. Modern machinery and productive methods precluded the continuance of such a system, and industry was frankly doubtful of its return. Moreover, greedy corporations had brought the term "apprentice" to mean little more than an ill-paid helper. But the economic need of better-trained mechanics showed that something had been lost in the passing of honest apprenticeship, and a new effort was thus forced upon our larger industries.

Second, that of development. Alteration and evolution of plan followed in this phase of progress. Management began to realize that the new apprenticeship contracts were a practical advance upon the old-time indenture—less rigid in their terms, more appealing to our young men, more adapted to modern shop conditions, and having desirable contacts with existing agencies of vocational education. These objections were desirable alike to employer and employee, but the specter of costs often appeared at the 'directors' board. The new immigration laws were a continued incentive to find a way of financing an apprentice plan. No longer could eager employment managers expect to meet their need of expert machinists from the trained output of European shops. But realization was also gaining ground that our young men could not expect forever to crowd into the congested ranks of the engineering professions, and that a larger proportion must necessarily find their careers in the expert trades. A dignified and remunerative future in industry must be displayed to schoolboys and their parents so that at least an intelligent choice could be made. Otherwise the high-school current would continue to carry on to college thousands who there would find themselves misplaced. It was felt too, that something must be done for the "eliminations," for that sixty per cent of engineering students who are dropped for one cause or another from our technical institutions and often are at loose ends with life.

And here we may be said to be entering the third phase of apprenticeship: that of application.

We see at the present time in our body social numerous agencies either coöperating or ready to coöperate with our manufacturers and other employers in the equipping of young men with education and shop experience sufficient to enable them to attain the limit of their individual abilities in the industrial life of the country. Enacted laws and public sentiment are and have always been strongly united upon one program—the rudimentary education of all children. The school system, in all its ramifications, is statistically increasing its annual output of the equivalent of the public-school eighth grade. Never before has the slogan "earn while you learn" had so wide an appeal. Trade schools, within and without the state control, are growing in number and size under popular demand for special instruction in preparation for industry. Part-time, evening, and continuation classes are helping boys and men to understand the groundwork and structure upon which their labor is founded and built.

The economic status as well as the personal wishes of the pupil and his parents will always have to determine the time of entry, within legal limits, of those entering industry. But it is freely conceded that no one can better justify his place on the city payrolls than the teacher in charge of vocational guidance. The lack of such guidance, here and further up the educational ladder, is what is responsible for most of our misfits. The picture should be presented to the boy at this critical time, whatever his age may be, of what he may expect to find along the path he is entering so that he may choose not only congenially but with some glimpse of his future earning power. With the conviction established that one may expect by proving his worth to add special knowledge to his schooling and thus earn promotion, there is bound to be less crowding into the colleges and technical schools.

The approach of the job from the school is necessarily a gradual or trickle process. It can never logically be made in group formation or, in other words, made to conform to the usual school graduation periods. Economic facts again determine this control—which is fortunate, as the demand for workers must ever be a fluctuating one. If a boy is so favored that he lives where his school can arrange for him to have some valuable shop training while he is adding practical subjects to his fundamental and academic studies, he may often enter his apprenticeship gradually with night-school instruction to satisfy his need for some specialized knowledge or to supply some conscious lack of information. In some communities it has proved best for apprentices to be sent under pay to the local vocational or trade school before actually entering the shops of their respective employers. Special merit in the training of apprentices unquestionably lies in that coöperative system which, under an independent supervisor, routes them through a group of shops for their factory experience. The apprentice to a small concern loses something through inability to see divergencies of shop practice and to note comparative workmanship and methods.

The larger corporations avoid this limitation by the diversity of their departmental organization, and to them the self-contained apprentice school is not only possible but desirable. These corporation schools are sometimes conducted independently of the city schools, taking boys with the equivalent of eighth-grade standing and providing general and vocational class work in conjunction with shop training. One enterprise now in process of organization proposes to undertake the responsibility of primary-school instruction to the end, presumably, that loyalty to the company be inculcated at an early age. Many of these apprentice schools are admirable in the thoroughness of their training, usually worked out in conjunction with the local educational institutions. They almost invariably certify their graduates as journeymen in a recognized trade, often stating on the certificates or diplomas the number of years required to qualify.

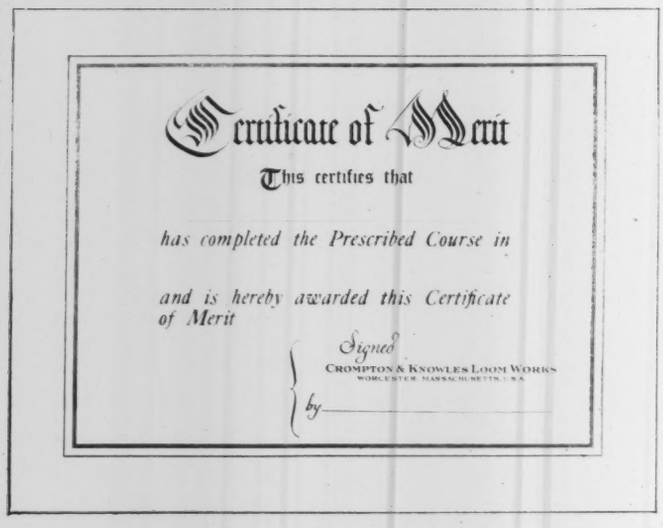
Some thirty-four of these forms have been compared. They range in size from 17 in. × 26½ in. down to pocketbook dimensions. Five of them only are issued on a time basis or for the completion of a prescribed course, but none of these are strictly in trade work. Of the other 29 certificates, 25 give a definite trade or vocational rating on a basis of proficiency (one with recommendation for employment), 2 give a professional rating, and the remaining 2 are issued on a statement of general proficiency. Each of these large companies has its definite objectives in apprentice training and is doing good work in industrial education. The only thing which may be urged upon all, in the interest of their graduates, is a greater uniformity in the training for each trade and a more explicit statement of accomplishment in the certificates awarded.

Another collection of certificates was made of private and public trade and vocational schools, the latter through the courtesy of the Federal Board for Vocational Education. It is interesting to note that of these, numbering 130 in all, by far the greater

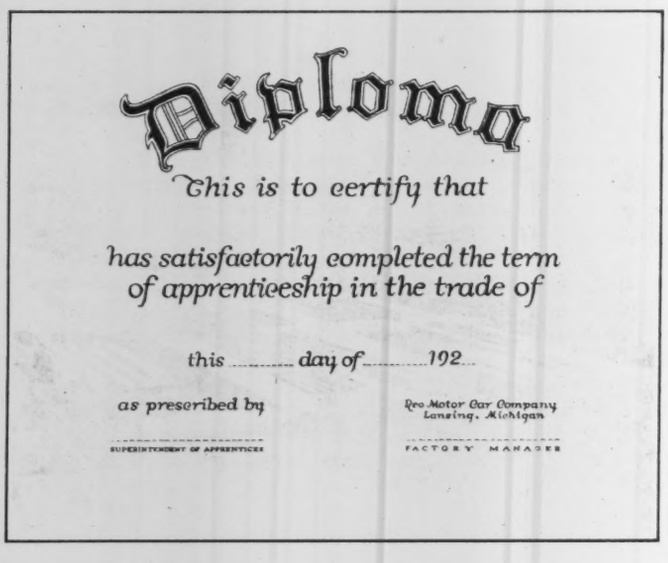
[1] Consulting Engineer. Mem. A.S.M.E.

Contributed by the Committee on Education and Training for the Industries for presentation at the Spring Meeting, White Sulphur Springs, W. Va., May 23 to 26, 1927, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 29 West 39th St., New York. All papers are subject to revision.

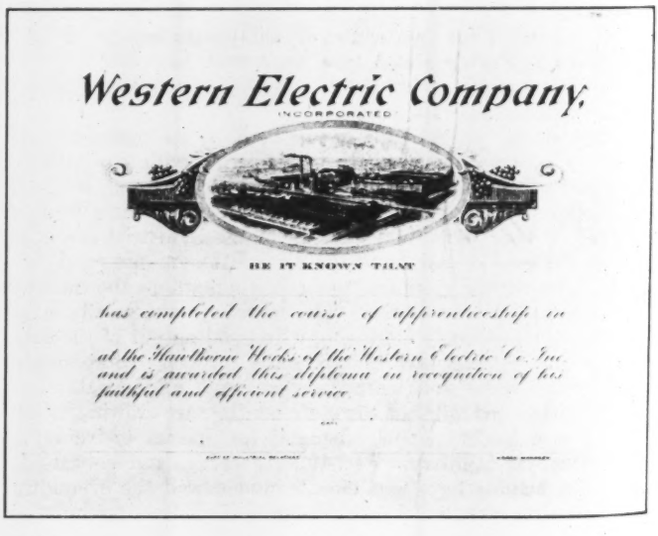
Be sure to bring this copy with you to the Spring Meeting



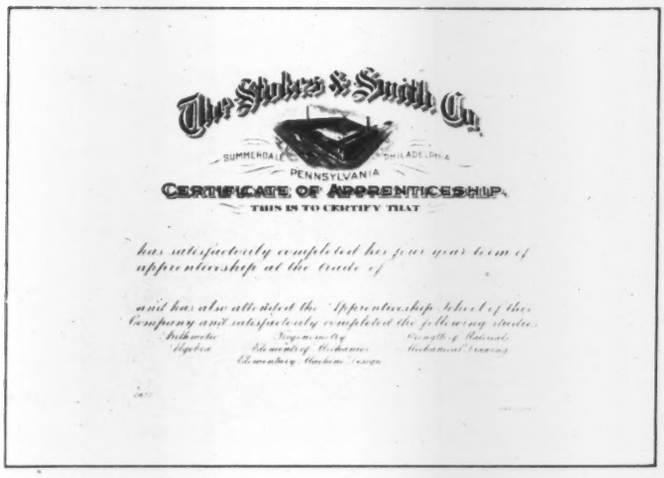
International Harvester Companies
Shop Schools
IHC
is awarded to Mr.
of
in recognition of his having creditably completed the four year course of work in the I. H. C. Shop School at
Works.



The Lakeside Press
R. R. Donnelley & Sons Company
Chicago Illinois
This is to Certify that
having completed the pre apprentice course of two years in the School for Apprentices and having served an apprenticeship of years in the of The Lakeside Press, is entitled to the rank of Journeyman with all the privileges of said position.
In Testimony Whereof, This diploma is awarded this day of 19 at Chicago Illinois, by the officers of the R. R. Donnelley & Sons Company.


Western Electric Company
Incorporated
Be it known that
has completed the course of apprenticeship in
at the Hawthorne Works of the Western Electric Co. Inc.
and is awarded this diploma in recognition of his faithful and efficient service.

Company Apprenticeship Certificates


The Stokes & Smith Co.

PENNSYLVANIA

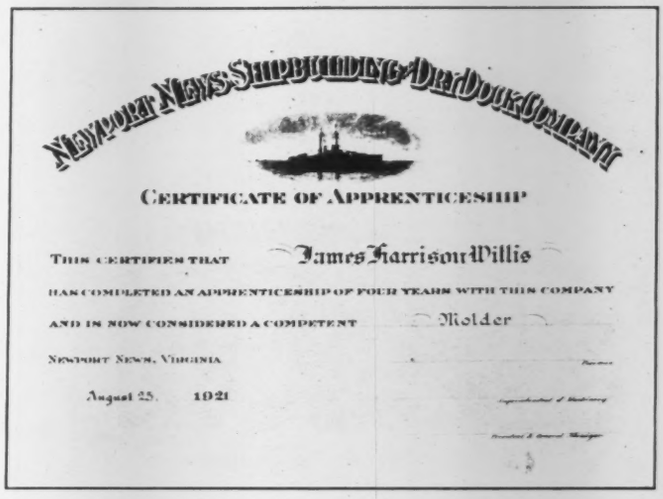
CERTIFICATE OF APPRENTICESHIP

THIS IS TO CERTIFY THAT

has satisfactorily completed his four year term of apprenticeship at the trade of

and has also attended the Apprenticeship School of the Company and satisfactorily completed the following studies:

Arithmetic — Algebra — Elementary Mechanics — Elementary Machine Design — Geometry & Rational — Mechanical Drawing


NEWPORT NEWS SHIPBUILDING & DRY DOCK COMPANY

CERTIFICATE OF APPRENTICESHIP

THIS CERTIFIES THAT James Harrison Willis

HAS COMPLETED AN APPRENTICESHIP OF FOUR YEARS WITH THIS COMPANY

AND IS NOW CONSIDERED A COMPETENT Molder

NEWPORT NEWS, VIRGINIA

August 25, 1921

Union Pacific System

Certificate of Apprenticeship

Has served an Apprenticeship in the shops of the

at , and is hereby declared qualified as a journeyman in the craft

and in evidence thereof, this certificate is issued at 19

COMMONWEALTH FELLOWSHIP SCHOOL

COMMONWEALTH STEEL CO.

Certificate of Attainment

This Certifies that has faithfully pursued the prescribed course in

in the Commonwealth Fellowship School for apprentices and other employees and has given satisfactory evidence of his completion of the course.

Granite City, Ill.,

Dated 192

Certificate of Apprenticeship

This Certificate issued by

declares that

has satisfactorily completed the prescribed Apprenticeship Course of years at and has received the requisite practical training to fit him as a journeyman.

In testimony thereof this Certificate is duly signed and issued this day of

OFFICE OF R. HOE & CO.

PRINTING PRESS MACHINE AND SAW MANUFACTURERS

NEW YORK WORKS — GRAND STREET

LONDON WORKS — MANSFIELD STREET, BOROUGH ROAD, LONDON, ENG.

CERTIFICATE OF APPRENTICESHIP

GRANTED TO

New York

This is to certify that has served his full term of apprenticeship in these works as from to

During years of his term he attended two Apprentice Schools in the evenings studying Mathematics, Spelling and Mechanical Drawing. His general rating is that of

R. HOE & CO.

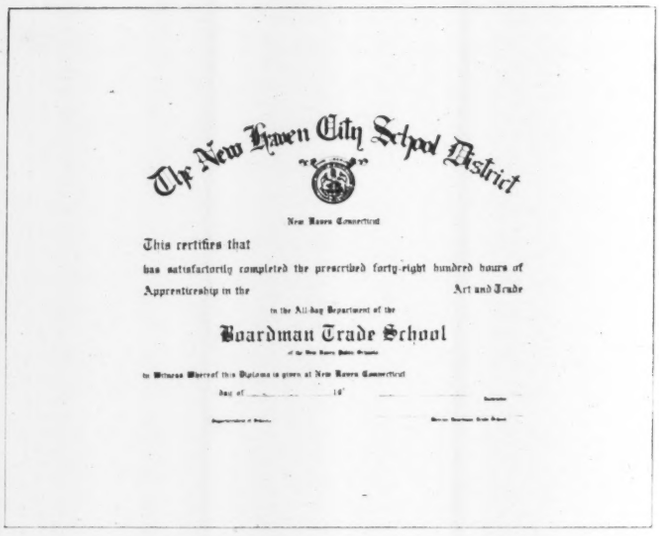
Company Apprenticeship Certificates

State of Connecticut

TRADE

APPRENTICE CERTIFICATE

THE STATE BOARD OF EDUCATION


The New Haven City School District

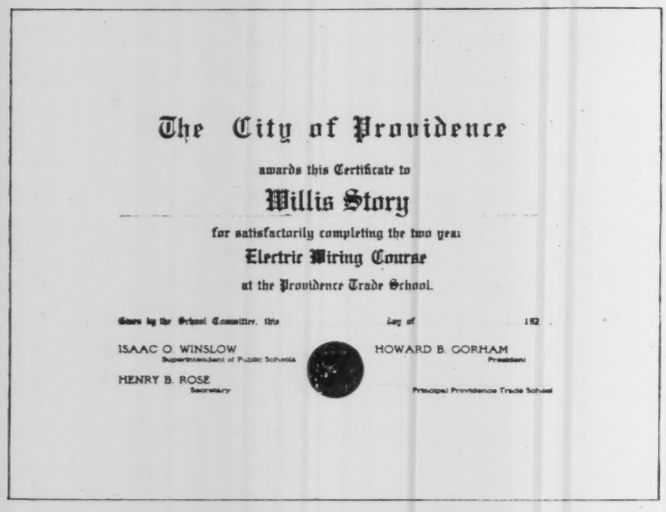
New Haven Connecticut

This certifies that

has satisfactorily completed the prescribed forty-eight hundred hours of Apprenticeship in the Art and Trade

in the All-day Department of the

Boardman Trade School


The City of Providence

awards this Certificate to

Willis Story

for satisfactorily completing the two year

Electric Wiring Course

at the Providence Trade School.

ISAAC O. WINSLOW
Superintendent of Public Schools

HOWARD B. GORHAM
President

HENRY B. ROSE
Secretary

Principal Providence Trade School

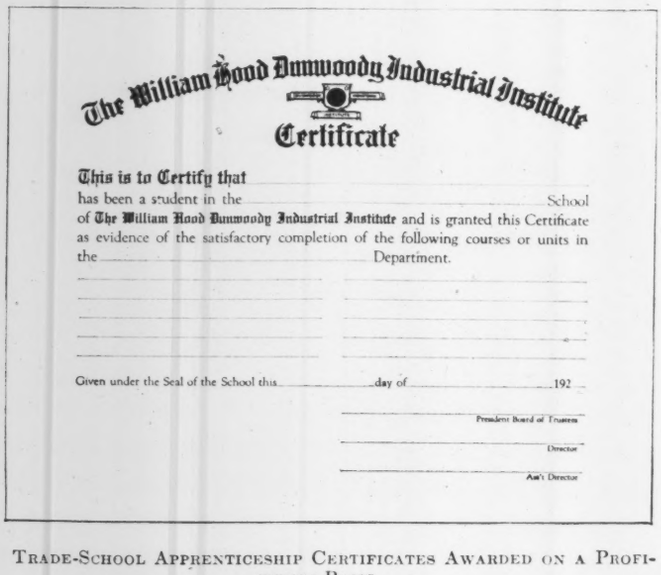
Joliet Township High School

This Certifies That

Having completed the two-year trade course in

is hereby declared a graduate of the two-year trade course and is entitled to this

CERTIFICATE


The William Hood Dunwoody Industrial Institute

Certificate

This is to Certify that

has been a student in the School of The William Hood Dunwoody Industrial Institute and is granted this Certificate as evidence of the satisfactory completion of the following courses or units in the Department.

Given under the Seal of the School this day of 192

President Board of Trustees

Director

Ass't Director

Trade-School Apprenticeship Certificates Awarded on a Proficiency Basis

part (114) are issued on a time basis—42 in preparation for trades and 72 for other vocational training. The remaining 16 of these schools not operated by industry give certificates stating the acquirement of proficiency: 5 in general terms, 7 with a trade rating, and 4 with recommendation for employment. Some of these trade schools have arranged with local manufacturers, to their mutual advantage and to the great benefit of the students, to accept in a more or less formal way relays of boys on the so-called coöperative or alternating plan. The reciprocal period

Apprentice Training School
College of Engineering
University of Tennessee
Knoxville

This Certifies that JOHN DOE of MARIETTA, GEORGIA has satisfactorily completed the prescribed course of study covering a period of three months, JANUARY 5, 1926 to MARCH 5, 1926, and is therefore recommended as an advanced apprentice in the trade of MARBLE SETTING

The Board of Managers
OF THE
ISAAC DELGADO CENTRAL TRADES SCHOOL
New Orleans, Louisiana
TO ALL WHOM IT MAY CONCERN
GREETINGS:
Be it known that the Board of Managers have been advised by the proper Faculty of the School that ____ has completed the course of study and training in the Trade of ____ and is hereby recommended for employment in that trade. DATE

The North Dakota
State School of Science
Wahpeton
THE TRADE SCHOOL
This Certifies that ____ has satisfactorily completed the regular Apprentice Course as prescribed by the Department with the following records:
and is entitled to this
Apprenticeship Certificate
In testimony whereof, ____

Cooperative Apprenticeship Certificates

of class room and shop work varies from five days to several months. The system has met the need felt in practical training for the vitalizing effect of productive competition, a condition never possible in the school shop no matter how well equipped it may be with modern machine tools. This is recognized in the use of the plan by schools whose graduates expect to enter business. It has been an active agency in calling the attention of the oncoming generation to the discrepancy between the industrial opportunities and the number of men able to fill them competently. The manufacturer has welcomed it, more or less enthusiastically, as a help in his problem of overcoming this widespread incompetence. The A.S.M.E. has members who are actively interested in both sides of this practical movement, which is gradually improving the quality of industrial workers.

It is, however, to the smaller manufacturing concerns themselves that the Society can with greater complacency offer its assistance. Having passed the phase of skepticism, an increasing number of works managers would like to approach their directors with some scheme of apprentice training, could they be assured that the plan would not be swamped by prohibitive costs. The hope of success lies in double coöperation. First, coöperation between a voluntary group of employers who subscribe necessary funds and organize a system of training with a supervision of apprentices; and then coöperation between this group and the existing vocational or trade schools of the district. There is nothing new in this plan. It is being demonstrated and needs only adaptation to local requirements and greater publicity in employment centers. It is equally applicable to the building, metal, textile, printing, and other trades. The scope of this paper cannot include illustrative and appreciative comment. The effort, however, has again been made to collect such certificates as have been jointly issued by schools and manufacturers. An analysis of these and of the plans which they represent seems to indicate that the fulcrum upon which the balanced system operates is the personality of the supervisor. It is through his display of tact, ability, and impartiality that the school is able to develop its vocational usefulness, that the boy is able to obtain the thorough training and experience which will steady his efforts at advancement, and that the employer is able to get the intelligent skill which he can afford to reward.

The collection of trade certificates is far from complete, as is inevitable in what would appear to be the first attempt of the sort. An indication of the interest felt in the subject may be seen in that but a very small percentage of the schools and manufacturers addressed failed to answer. It was also interesting to find that a considerable number of employers had not yet compiled any certificate or were about to revise those hitherto used. Thus, in order to draw some value from the study of these certificates, the result is here presented of the attempt to make a composite form, embodying the essentials for any double cooperative district apprentice system (see 2nd column, page 6).

The size suggested is that of ordinary business letter paper, in place of the larger diplomas intended for framing. While these

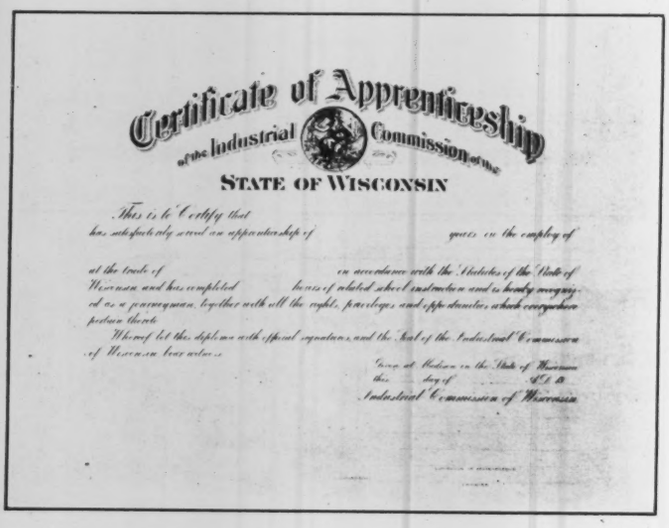
THE CLEVELAND METAL PRODUCTS COMPANY
AND THE
CLEVELAND BOARD OF EDUCATION

This is to Certify that
has successfully completed his **Four Year Apprentice and Vocational Course of Instruction**, in Shop Practice, Mechanical Drawing and Shop Mathematics.

In testimony whereof this diploma is awarded this

Vice-President, Cleveland Metal Products Co.

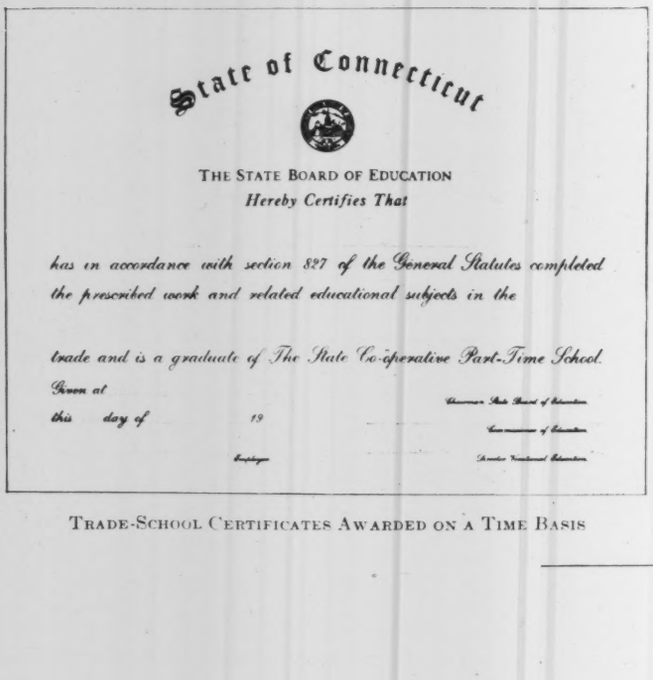
Director of Vocational Education, Cleveland Public Schools



Certificate of Apprenticeship
of the Industrial Commission of the
STATE OF WISCONSIN

This is to Certify that
has satisfactorily served an apprenticeship of years in the employ of
at the trade of in accordance with the Statutes of the State of
Wisconsin and has completed hours of related school instruction and is hereby recognized as a journeyman, together with all the rights, privileges and opportunities which accompany
persons thereto.
Whereof let the diploma with official signatures and the Seal of the Industrial Commission of Wisconsin bear witness.

Given at Madison in the State of Wisconsin
this day of A. D. 19
Industrial Commission of Wisconsin



State of Connecticut

THE STATE BOARD OF EDUCATION
Hereby Certifies That

has in accordance with section 827 of the General Statutes completed the prescribed work and related educational subjects in the

trade and is a graduate of The State Co-operative Part-Time School.

Given at
this day of 19

Chairman State Board of Education

Commissioner of Education

Employer

Director Vocational Education

TRADE-SCHOOL CERTIFICATES AWARDED ON A TIME BASIS

APPRENTICESHIP CERTIFICATE

The Public Schools of ..... (and the ..... State Board of Education)

in cooperation with

.....
(Name of employer)

jointly certify that

.....
(Name of apprentice)

has satisfactorily completed a ..... year course in Shop Practice, Mechanical Drawing, Shop Mathematics, Business English, and related educational subjects at .....

and a ..... year course of practical training in the ..... of

.....

with sufficient diligence and proficiency as to qualify as JOURNEYMAN in the trade of ..... and is recommended for employment in that trade.

His school work has also consisted of

Common School through the ..... grade;

High School .....

Vocational School .....

Continuation School .....

Night School .....

Part-time Day School .....

Full-time Trade School .....

Polytechnic School .....

Signed ..... 19 .
(Place) (Date)

....
(Employer) (Director of Vocational Education)

.....
(District Supervisor of Apprentices)

(Other agencies coöperating in the apprentice training program may be added)

can, if desired, be also awarded, the practical need of the young journeyman is for a certificate which can be presented to any employment department and which will be recognized as a standardized summary of his qualifications for the job he wishes to get. No manufacturer desires to maintain a vestibule school. It is sometimes an irksome necessity, but the facts certified on the appended form should appreciably reduce that necessity. From the standpoint of the applicant, his interview questionnaire is answered authoritatively, and he has but to make good upon the record of his training.

The General Motors Institute of Technology

Particulars of an Educational Program Designed to Provide the General Motors Corporation with a Supply of Highly Skilled Service Men and Trained Workers for the Plants, and to Furnish a Broad Technical Training for Its Future Executives

By ALBERT SOBEY,[1] FLINT, MICH.

THE General Motors Institute of Technology, which has recently been established in Flint, Michigan, is the outgrowth of a very interesting experiment in industrial relations, and in its present setting is unique in its organization and relationships as well as in its program and activities. It is in reality a new name representing a wider application of an industrial-education and training program which has been developed in Flint for the industries of that city over the past eight years under somewhat unusual conditions.

Years ago, under the leadership of Dallas Dort, there was established in the industrial life of Flint a policy of coöperation and mutual effort on the part of the managements and employees of the industries of the city that has been unusually successful in maintaining the morale of Flint's industrial life, and a big factor in the remarkable growth that this city has enjoyed.

This policy was established some twenty years ago in connection with the problem of insurance for the employees of industries, and resulted in the organization of a mutual-benefit association for those employed in all the industries of the city; this became the center of an employees' club, with recreational as well as insurance features, controlled and operated by the workers and enjoying the endorsement and backing of the managements of the factories.

With the growth of the automobile industry and the coming of the World War, industrial unrest, labor turnover, and the other problems common to that period brought an extension of this policy, finally resulting in what is now known as the Industrial Mutual Association, an organization of all the employees of the city, with over 30,000 members and an active program covering almost every phase of life related to industry, centralized in an employees' club of some 60,000 sq. ft. of floor space in one of the finest buildings in the heart of the city, and operated and controlled by a board of eighty trustees elected by the employees of the plants of the city. It is unquestionably one of the most successful organizations of industrial employees in the country, and a striking example of the practical value of coöperation.

Early Development of Educational Program at Flint, Michigan

It was in connection with this employees' organization that the educational program which has now developed into the General Motors Institute of Technology, had its origin. Realizing the growing importance of a definitely organized program of education and training to meet the personnel problems involved in the constantly increasing specialization of modern industry, the managements of Flint industries adopted the policy of coöperating with their employees' organization in the development of a program for their plants. While thus recognized as an employees' program, the work enlisted the support of large numbers of men from practically every stratum of Flint's industrial life—managements, executives, technical-staff experts, and workmen. The result has been a comprehensive program which has been rapidly extended to cover all the major fields of training connected with the automobile industries, and which has constantly grown in interest and attendance. This development has not been one-sided—organized for one group only—but has been developed with full consideration for the interests of all factions involved: employees, the members of their families, and the plants.

As we all know, changes in industrial organizations and methods during the past generation have greatly lessened opportunity for development in the regular processes of industry, so that the matter of training employees for positions requiring skill and ability has become a vital problem to the industrial executive, involving the morale and effectiveness of his entire organization. The old apprentice system, by means of which industry has trained its people for its more highly skilled trades from the beginning of the machine era, has become practically a thing of the past, and specialization has been carried to such a point that the regular processes of industry no longer offer much opportunity for a well-rounded experience and training to qualify men for the more responsible types of positions.

Modern industry, in fact, has been likened to a tall building of many stories without elevators or stairs. On each floor, beginning with the simpler tasks on the ground level, are various classifications of specialized jobs extending up to the highest executive at the top. It is easy to qualify for entry at the ground floor, but industry needs a constant supply of qualified employees for the floors above. Specialized as it is, it lacks the means to train and prepare its employees for advancement to the floors above. Industrial leaders, therefore, are turning to special organizations to provide the means of training—ladders by which employees can climb to more highly skilled and more responsible positions above.

This does not mean that "university of hard knocks" has ceased to function. The practical school of experience still plays an important part in industrial training, but unless it is correlated, directed, and supplemented with needed technical instruction, it is too haphazard and wasteful of time, effort, and human energy.

It might be likened to attempting to crawl up the walls of a building from floor to floor. A factory manager recently explained it in this way. "I came up from the ranks. It has taken me twenty-five years to gain the knowledge and grasp of fundamentals to make a success of this job. Starting now in a similar way but under modern factory conditions, it perhaps would take fifty years. This is too wasteful. It should be possible to do it in ten years, possibly in five."

When I was a boy serving a four-year apprenticeship in the shop, for example, the master mechanic would frequently put on overalls and spend a half or even a whole day at a time working with me, stating, in his own words, "I am going to 'learn' you all there is to know about the business." In contrast the other evening a young man of fine appearance came to the office and presented the following case. "Mr. Sobey, I have been working for the Buick Motor Company for six years. I have had a high-school education and believe I have normal intelligence and ability. I am ambitious to get somewhere in life and work myself up to a responsible position for the sake of my family and myself. After six years I find I am getting nowhere and can see no chance ahead in the plant. What can you do for me in school to help me to advancement?" It happened that the day before this visit we received a call from the superintendent of the plant in which the young man was working, asking for some one from our night-school program qualified to fill a junior-executive type of position in that plant—just the chance the young man was looking for. The opportunity was there, the Buick Motor Company was interested in its employees and in need of qualified people for advanced positions, but the nature of the young man's work did not give him an opportunity to prepare for it—not in six years. This case illustrates the situation faced by large numbers of ambitious employees of modern industry.

Flint has attempted to meet the problem thus presented through a coöperative effort of management and men in the development of an educational and training program directly related to the needs of Flint industries and her employees. The needs of the employees have been met, largely through a spare-time course of training, and

[1] Director, General Motors Institute of Technology.

Contributed by the Committee on Education and Training for the Industries for presentation at the Spring Meeting, White Sulphur Springs, W. Va., May 23 to 26, 1927, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

Be sure to bring this copy with you to the Spring Meeting

the needs of the plants, in so far as they are not met by the spare-time program, in short, intensive, full-time and in coöperative courses.

SPARE-TIME PROGRAM FOR PRESENT EMPLOYEES

It is natural that the development of the spare-time program for present employees should be given first consideration. Investigation of the plant conditions at the beginning of the work showed here and there throughout the entire system groups of men ambitious for advancement along the lines on which they were then engaged, with little opportunity in the specialized nature of their positions to acquire the training needed for such advancement. Here and there were other groups of men placed in positions for which they were not well adapted, and whose interest lay in other

INDUSTRIAL ENGINEERING COURSES

FIRST YEAR

Fall Term.		
D-1	Mechanical Drawing—1st term	4 hrs. per week
B-3	Geometry	2 hrs. per week
Winter Term.		
D-2	Mechanical Drawing—2nd term	4 hrs. per week
B-2b	Advanced Algebra—1st term	2 hrs. per week
Spring Term.		
D-3	Mechanical Drawing—3rd term	4 hrs. per week
B-2c	Advanced Algebra—2nd term	2 hrs. per week
Summer Term. (8 weeks)		
W-21	Engineering Pattern Making & Foundry Practice	6 hrs. per week

SECOND YEAR

Fall Term.		
N-1	Practical Physics	4 hrs. per week
B-4	Trigonometry	2 hrs. per week
Winter Term.		
C-1	Practical Chemistry	4 hrs. per week
N-1e	Electricity	2 hrs. per week
Spring Term.		
C-2	Practical Chemistry—2nd term	4 hrs. per week
N-2	Elementary Practical Mechanics	2 hrs. per week
Summer Term. (8 weeks)		
S-22e	Engineering Machine Shop Practice	6 hrs. per week

FIG. 1 FIRST AND SECOND YEARS OF SPARE-TIME INDUSTRIAL ENGINEERING COURSES

(Note the suggested sequences of unit courses. The program is so flexible that a student may start at any stage, depending upon his previous education and experience, and make the arrangement of unit courses that best meets his individual needs.)

channels, with little opportunity to make the transfer. Here were groups of men growing older with the end of their productive years on the present heavier jobs immediately before them, but whose productive period could be increased if arrangements could be made to train them for lighter types of jobs involving some skill. Here also were foremen ambitious to climb the ladder to managerial positions, and men of promise ambitious to prepare for foremanship responsibilities. There were groups of men in the plants whose greatest possibilities were in office types of position, and men in the office who should be in the plant. In a word, there were a large number of diversified groups neither happy nor contented on the present jobs, but ambitious for advancement or transfer, and who would gladly accept opportunity for special training which would help them qualify for a broader, more satisfying life if arrangements could be made.

To meet this situation a spare-time program of some 200 unit courses has been developed covering practically every phase of the automobile industry. These unit courses are arranged in major sequences of training for the more important branches of work in the industry, and designed to meet the needs of the average student. In practice they are arranged very much as is a cafeteria, the various units being spread out before the ambitious employee in the order suggested for the average man, but in any individual case they can be selected in almost any order. Thus, it is possible for an ambitious employee to select the unit of training that he needs for his next step in advance and build upon it a very flexible program that can readily be adjusted to meet his changing fortunes.

The way in which these short unit courses are grouped together in major sequences is illustrated in Figs. 1 and 2 showing the first two years of industrial engineering courses, and the third and fourth years of the specialized industrial tool engineering course. The entire spare-time program includes seven suggested sequences similar to this tool engineering course, and also, a six-year sequence of executive training, a foremanship program, three one-year sequences in accounting and business, twelve trade-training programs ranging from one to four years in length, and also special job-training courses along various lines of work of the semi-skilled type and a number of more elementary courses to give opportunity to men of limited education to prepare for entrance to the major courses referred to. A program of courses for girls is also included.

COURSES INCLUDED IN SPARE-TIME PROGRAM

With this spare-time program organized and in operation, the next step was to formulate a program to more fully meet the needs of the plants. Three courses are included in this program as now organized:

1 The Buick Authorized Service Course
2 The Technical Trades Course
3 The Coöperative Engineering Course.

The Service Course is designed to meet the needs recognized by every one for more highly skilled service men for the service organizations of the country. It is conducted upon the sound principle of placing emphasis in the instruction upon the funda-

INDUSTRIAL TOOL ENGINEERING COURSE

FIRST AND SECOND YEARS

Fall Term.		
D-5	Elements of Engineering Design	4 hrs. per week
N-3	Elementary Mechanics (Advanced)	2 hrs. per week
Winter Term.		
D-6	Elements of Engineering Design—2nd term	4 hrs. per week
N-4	Strength of Materials	2 hrs. per week
Spring Term.		
D-8	Tool Design	4 hrs. per week
M-11	Metals and Their Heat Treatment	2 hrs. per week
Summer Term. (8 weeks)		
S-23e	Engineering Machine Shop Practice	6 hrs. per week

FOURTH YEAR

Fall Term.		
D-9	Tool Design—2nd term	4 hrs. per week
R-2	Inspection Tools and Methods	2 hrs. per week
Winter Term.		
D-10	Tool Design—3rd term	4 hrs. per week
F-1a	Engineering English	2 hrs. per week
Spring Term.		
D-10a	Tool Design—4th term	4 hrs. per week
F-16	Engineering English	2 hrs. per week

FIG. 2 THIRD AND FOURTH YEARS IN "TOOL ENGINEERING"

(This is one of seven similar programs in specialized branches of engineering related to the automobile industry.)

mental principles of car construction, and modern methods of maintenance and repair and their application to just one set of conditions—the Buick car. As a result, the Buick Service Course has been strikingly successful and its graduates are now filling responsible positions in every state of the Union and in seventeen foreign countries. That it has been adopted by the General Motors Export Company for training its service men and giving a technical training to its salesmen, is evidence of its thorough, practical nature and the correctness of the principles of instruction utilized.

The Technical Trades Course is designed to meet the needs of the industry for highly skilled craftsmen in tool and die making. It is organized on the coöperative plan of alternate periods of four weeks in school and four weeks of practical training in the plant, and offers to boys 16 to 18 years of age, who have had a tenth-grade education or its equivalent, an opportunity to obtain a trade training and a thorough related technical instruction of high-school grade, but of a specialized technical and trade type.

The Coöperative Engineering Course is a part of a definite program to meet the growing problem of foremen and executives in modern industries. As specialization has increased, the demands

upon the workmen have decreased, but there has been a corresponding and steady increase in the demands upon foremen and executives. This constitutes one of the major problems of industry today, and to meet it in Flint, three distinct programs were organized.

For men already in foremanship positions or serving as junior executives, a number of unit courses were arranged covering the principles of foremanship, the elements of management, and practical economics. The training for future executive material was organized in two separate programs, one for young men of promise already employed in the industries, and the other for those of the right type coming out of high school.

A very interesting situation was met with in the organization of foremanship program. The first courses were offered as a part of the regular spare-time program, which of course is participated in primarily by employees below the rank of foremen. The courses were very carefully brought to the attention of every foreman in the city, but with no response, and an investigation showed a reluctance on the part of the foremen to go into what they considered an educational competition with their men. Steps were then taken for the organization of foremen's clubs in all the plants of the city and for combining them into a central association with a central committee composed of representatives from each plant group. The educational program was then organized as a program of this foremen's organization, with the result that it met with enthusiastic response and has been extended, largely because of the demands of foremen for continuing courses. After several years of operation under these auspices, it was made a definite department of the regular school program, and has continued to function successfully since that time.

In the development of the executive training for young men of promise already employed and who, for economic or other reasons, must work on a regular job, a six-year program of spare-time courses was organized. This covers, on a night-school basis, two years of the fundamentals of engineering and two years of what we call junior-executive training in such subjects as the historical development of industry; the elements of economics, accounting, and costs; and personal-development courses in effective speaking, personal efficiency, and salesmanship. This is followed, for men who successfully qualify for foremanship or junior-executive positions, by two additional years of senior-executive courses covering foremanship and other courses directly related to industrial organization and management.

The Coöperative Engineering Course

The Coöperative Engineering Course was organized to supplement this source of supply of future executives with a group of young men more broadly trained technically. The course is a four-year program of college grade, for admission to which high-school graduation or its equivalent is required and also evidence of good character and some potential leadership ability. The program, which is conducted on the coöperative plan, with alternate periods of four or eight weeks in the Institute and corresponding periods in the industry, is in reality fourfold in nature. During the Institute periods the training may be classified under two heads: (*a*) fundamental engineering subjects that are common to all college engineering courses, and (*b*) specialized training in the principles of industrial organization and management. During the work periods the student is given a directed practical experience in industrial operations, which also gives him contact with and a knowledge of industrial men.

This Coöperative Course is illustrated in Fig. 3, which indicates the fourfold nature of the training that is offered. It also indicates the relationship between the various courses, which are arranged in horizontal sequences. This course, and in fact the entire foremanship and executive program, has been developed with the active coöperation of an advisory committee of leading executives of Flint who began their work in the early stages of foremen's organization and have taken a very active interest in all features of the development. Their work has been a big factor in making the program of real practical value, closely related to the problems of industry.

GENERAL MOTORS INSTITUTE OF TECHNOLOGY

CO-OPERATIVE ENGINEERING COURSE

ACTIVITY PERIODS	1ST YEAR	2ND YEAR	3RD YEAR	4TH YEAR	WORK PERIODS
FUNDAMENTALS OF ENGINEERING	ENGINEERING SHOP PRACTICE	ENGINEERING SHOPPRACT.			DIRECTED PRACTICAL EXPERIENCE
	ENGINEERING DRAWING	ELEMENTS OF DESIGN	TOOL DESIGN		
	COLLEGE ALGEBRA-TRIGONOMETRY	ANALYTIC GEOM. CALCULUS	MECHANICS OF ENGINEERING		
	CHEMISTRY. PHYSICS. CHEM. OF MATLS		HEAT TREAT. & TESTING MATLS	ELECTRICITY	
	ENGLISH COMPOSITION		ENGLISH LITERATURE	ENGINEERING ENGLISH	
		EFFECTIVE SPEAKING	PERSONAL EFF. SALESMANSHIP	BUSINESS LAW	
INDUSTRIAL ORGANIZATION AND MANAGEMENT	PRIN. AUTO CONST. IND. HISTORY	ECON. ORG. INDUST. ORG.	MFG METHODS RTING & PLANT LYT.	INDUSTRIAL MANAGEMENT	CONTACT WITH AND KNOWLEDGE OF MEN.
			ECONOMICS ACCOUNTING	COSTS & COSTS CONTROL STATISTICS BUSINESS CYCLES	
			PSYCHOLOGY	APPLD PSYCH. & BAROMETERS	
			LABOR PROBLEMS	INCENTIVES TIME STDY RATE SETTING MANAGEMENT PROBLEMS	

FIG. 3 PROGRAM OF THE COÖPERATIVE ENGINEERING COURSE

(This is in reality a management engineering course with a close correlation of training in the Institute with actual practice in the industry with major executives advising with the Institute staff regarding the training program.)

The purpose of the Coöperative Engineering Course is to prepare young men of the right type for future executive positions. It is really a management-engineering course with a close correlation of training and practice through connection of the Institute with the factory system. Not later than the junior year, and in many cases earlier, students are placed under the direction of some executive of a major division of our industries and are recognized as his men. From this point on he advises with the Institute staff regarding their training program with a view to having it of such a practical nature as to make the men most valuable for his own organization upon completion of their training. In many cases the courses are so conducted as to give direct contact and experience with corresponding factory problems in the Flint units, with factory executives coöperating with the regular instruction staff to make the instruction as vital and up to date as possible.

The school grew very rapidly. The students and factory employees, recognizing it as their own institution, entered into its training confidently and in large numbers; in fact, this year our spare-time enrolment included 4 per cent of the employed personnel of our factories. Regular monthly reports to the plants kept the factory executives informed of the progress being made in school by their employees, and led to recognition and advancement in large numbers of cases as the men prepared themselves for advancement in their school work. The result was a record of growth (shown in Fig. 4) which last year brought the Institute to a place where the limit of its facilities was reached.

Broadening the Scope of the Work Begun at Flint

At this point the late H. H. Bassett, president of the Buick Motor Company, became personally interested in the problem of securing adequate quarters for the continued development of the school, and after a study of the situation decided to ask the

Executive Committee of the General Motors Corporation to provide a building and grounds in order to make possible the continued growth of the program in order to meet the growing needs of the Flint units of the corporation. This was done at the May meeting, held in Flint in connection with a visit of investigation and inspection of the school. As a result it was decided not only

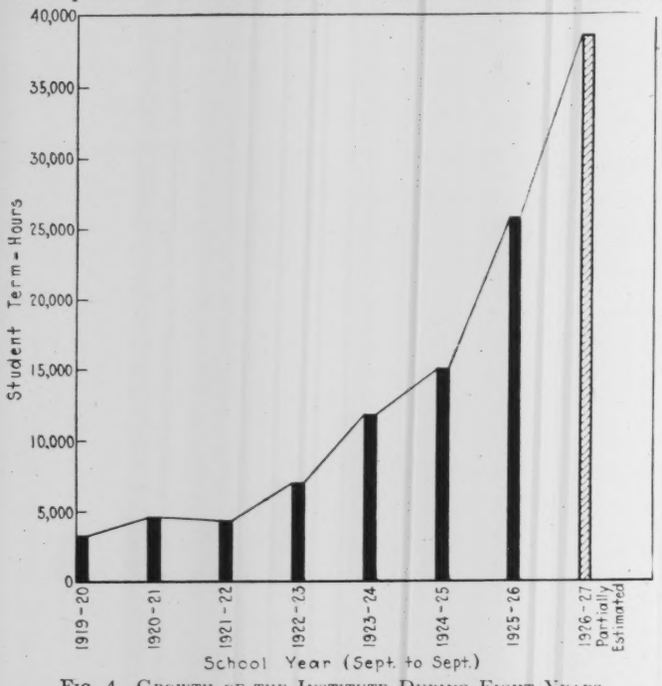


FIG. 4 GROWTH OF THE INSTITUTE DURING EIGHT YEARS
(The growth of the Institute is charted in student term-hours. Last year it reached a rate which brought the institution to the limit of its facilities, and under the leadership of the late H. H. Bassett led to the movement for the new building and new campus provided by the General Motors Corporation.)

to provide the building and campus required by the Flint units, but also to change the name to the General Motors Institute of Technology and provide for the extension of the program to all units of the corporation outside of Flint as well as in the city. The size of the campus as originally contemplated was tripled to provide room for expansion.

The building has now been completed and the Institute is functioning there. It has approximately 65,000 sq. ft. of floor space on three floors. The first floor is devoted to the auto mechanics laboratory, machine shop and tool room, and a service station; the second floor to general offices, a library, physical, chemical and metallurgical and electrical laboratories, and a large assembly hall and gymnasium, while the third floor is devoted to drawing, class, and study rooms. Figs. 5 to 8 show various views of the building layout.

The general policy upon which the program of the General Motors Institute of Technology is now being worked out follows very much the same lines as those which have proved successful in connection with the Flint program. Courses, such as the Coöperative Engineering Course and Service Courses, which can be centralized to an advantage at the Institute will be given there. In fact, students have been entered in the Coöperative Engineering Course by all the car-producing units of the corporation, and by the units at Dayton this year, and arrangements are now being made for all the other units of the corporation to enter students in this program at the beginning of the next school year in September. Plans are also being made for the extension of the service course to the other units of the corporation.

It is also planned to have the Institute perform a centralizing function for the training activities of all the units of the corporation, which possibly will be even more significant than its local program. In this connection the results of any training development or research conducted at the Institute will be made available to all units of the corporation on what may be termed an extension basis. The results obtained from the use of these programs or any local programs developed by the units themselves will be reported back to the central institution where they will be analyzed and put in such form as to be of value to the other units. In this way it is planned to develop an extension program of training that will cover the major personnel-development problems of the corporation in a form that reflects the combined experience of the entire organization.

This work was begun during this year in connection with foremanship development. The foremanship courses of the Institute have been used in connection with the training of foremen at the Delco Remy Company, Anderson, Ind., Harrison Radiator Corporation, Lockport, N. Y., and the Olds Motor Works, Lansing, Mich. Some four hundred foremen have been included in this extension program. They were organized in sixteen different classes and suggestions that have arisen from the experiences of these men in the course are now being received at the Institute and analyzed with a view to giving them effect in connection with changes and improvements in the program for next year.

The General Motors Institute of Technology is thus a somewhat unique institution—an outgrowth of a program resulting from the coöperative effort of all elements of Flint's industrial life under employee auspices, it now combines with these advantages the endorsement and backing of the General Motors Corporation and the extension of its program to the 150,000 employees of the corporation in all parts of the country. It represents a company-wide effort to accomplish real constructive results for the employees and for the corporation in man-power development and human relations.

FIG. 5 VIEW OF THE NEW BUILDING OF THE GENERAL MOTORS INSTITUTE OF TECHNOLOGY

CHARACTER OF INDUSTRY OF FUTURE DEPENDENT ON CHARACTER OF EDUCATION GIVEN ITS WORKERS

It has been said that the character of industry in the future will be determined by the character of the education given to its workers regardless of whether this education be organized or unorganized, and it is not unlikely that the preservation of our civilization will depend upon these developments. Whether industry contributes effectively to the common welfare or is wasteful of human life and energy, whether it becomes a tremendous constructive agency or a menance to our civilization, depends in large part upon the education provided for it.

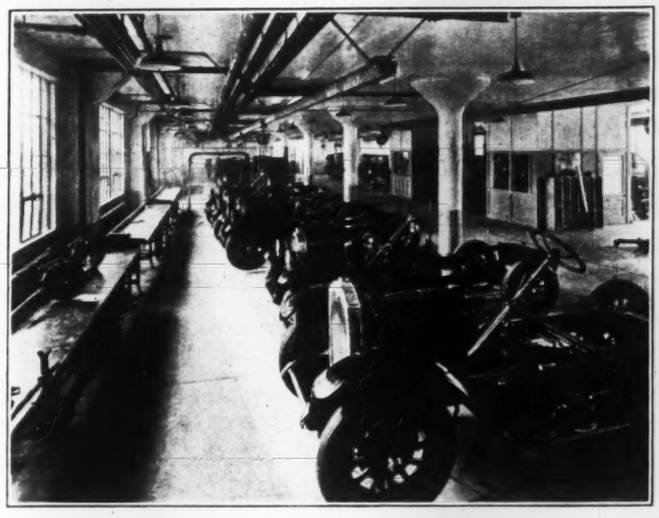


FIG. 6 SECTION OF AUTO MECHANICS LABORATORY

(Instruction in auto mechanics and automotive service has been specialized so far in the Buick product. Note the complete equipment for this work. It is planned to extend this program to other General Motors products.)

FIG. 8 REAR-AXLE AND TRANSMISSION DEPARTMENT

(Note section of machine shop and heat-treatment laboratory in the background. The machine shop occupies 10,000 sq. ft. of floor space and is equipped with fifty motor-driven machine tools.)

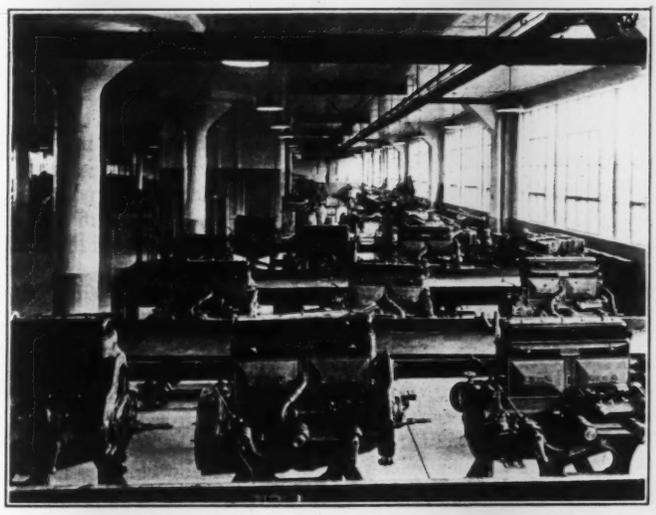


FIG. 7 ANOTHER SECTION OF AUTO MECHANICS LABORATORY

There certainly is need in our modern industry for an educational and training program to overcome the handicaps of specialization and provide for the development of its employees, increase their efficiency in industrial operations, and prepare them for advancement. The Flint program has accomplished very definite results in this connection. Hundreds of men have improved their position in life and are working their way up as a result of its program. These men range all the way from the young man just beginning his climb, to the man, who after seven years of training in the spare-time program, has worked himself up to a $10,000-a-year position.

There is, however, an even greater need from a broad social standpoint for a program which, in addition to these individual results, will provide industrial leadership of broad vision, promote better industrial relations, and bring a fuller understanding on the part of industrial people of the functions of industry as they relate themselves to life and the common welfare. There is a growing recognition that industry exists not for individual profit alone, but to serve owner, manager, workman, customer, and public. If industry is to effectively function in this regard, the ideal must not only be caught by the employer and manager, but in a measure it must also be disseminated through rank and file of the employees. This is a problem that can enlist the interest and constructive effort of all the various elements involved in the industrial organization.

It is to provide such an educational and training program and to contribute toward such results for the General Motors Corporation and for its employed personnel, that the General Motors Institute of Technology has been established. Its purpose is to serve this world-wide organization in developing its industrial man power and leadership, in broadening the understanding of its people regarding the functions of industry, and in the promoting right human relationships.

Arc Welding

Advantages of Arc-Welded Steel Parts over Those of Cast Iron or of Riveted Construction—Savings in Cost Effected by Replacing Iron Castings with Arc-Welded Steel Parts—Inconsistency of Permitting Arc Welding in Superheaters and Steam Piping and Forbidding It in Boilers, Etc.

By J. F. LINCOLN,[1] CLEVELAND, OHIO

ARC welding is a comparatively old process—that is, the first patents in connection with it were taken out more than thirty years ago. However, its use commercially, on a large scale, is comparatively recent.

Undoubtedly the repair of the interned German ships in the harbors of the United States, during the late war, brought arc welding to the attention of the American people a good deal more forcefully than any other application which has ever been made. The repair of these ships as we see it now was a comparatively simple operation, and while it undoubtedly saved a considerable amount of money and time for the American Government and increased materially the efficiency of the American intervention in the war, nevertheless the actual use of welding in this case and in all other repair cases must be of secondary importance.

While the use of arc welding for repairs undoubtedly saves this country in excess of one hundred million dollars per year, by far its greatest value in the future is going to be in the replacement of riveted steel and castings with arc-welded steel. The fundamental reason why this replacement will come is twofold.

Advantages of Arc-Welded Steel Parts over Riveted or Cast Iron Parts

Compared to riveting, arc-welded steel will give the full strength of the joint, while rivets will give but from 50 to 80 per cent of the sheet strength when the riveted joint is made in the usual commercial manner. This means, therefore, that any structure which is made of riveted steel and depends on the strength of the joint for its efficiency must have from 30 to 100 per cent more steel in it than would be necessary were an arc-welded joint used. What this would amount to in the aggregate is a stupendous sum, as can be well appreciated.

However, there is still another application which is of very much greater importance, namely, the employment of arc-welded steel in place of castings of all kinds. The fundamental principles involved in this substitution are as follows:

Cast iron has a tensile strength of approximately 10,000 lb. per sq. in.; that of welded steel is approximately 50,000 lb.

Cast iron has a modulus of elasticity of 12,000,000; that of steel is 30,000,000.

Castings on an average cost 6 cents per lb.; rolled steel, 2 cents.

The factor of safety necessary for the same degree of safety will be half as great with steel as with cast iron. Therefore, if a pressure vessel in which the strength of the shell is the limiting factor is made from cast iron, it will cost for the same degree of safety approximately twenty-five times as much as if made from rolled steel; if a base or any structure which is used for resisting bending is to be made, the cost will be but approximately 15 per cent as much if made from structural steel as if made from castings.

When the total amount of this saving is considered, it is easy to see that the sums involved run into figures that are enormous even to the American people who are accustomed to speaking glibly of billions.

Regarding the problems which are involved in taking advantage of these fundamental principles, it is quite evident that if the plan were merely a matter of replacing one metal with another there would be no doubt about its immediate adoption. But it is not nearly so simple as that; one great virtue of the casting is the fact that it can be made into any shape with very great ease, while, rolled steel must of necessity (from its very method of manufacture) be made in certain definite shapes and those only.

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These shapes can of course be so made that they will be the most efficient for the kind of work they are designed to do, and if they can be adopted directly without change they will do any work better because of this shape than would be possible with most cast shapes, entirely aside from their greater strength and stiffness.

Possibilities of Redesign of Cast into Arc-Welded Parts of Enormous Importance

However, there are a great many cases where it is necessary to make a structure which is radically different from any shape directly procurable from the steel mills, and therefore the possibilities of the designing engineer in redesigning cast structures into those using welded steel are of enormous importance. In fact, the designer of the future who can apply this new tool to best advantage will be an outstanding engineer.

The importance of this seems so great to the company with which the author is connected, that it has offered three prizes of $10,000, $5000, and $2500, respectively, for the best applications of arc welding to the redesign of present-day structures, hoping in this way to get the brains of the country at work considering the proposition more closely, more intelligently, and more thoroughly, so that the savings which such redesign will afford may be brought about in the shortest possible time.

One very definite matter which is holding back the universal acceptance of arc welding as an economical producing tool is the fact that habit so largely controls our minds. Mankind in general accepts without question the theory that if a precedent has been established, it must of necessity be right and should continue to be followed. Of course, all laws are based on this fundamental principle and perhaps that is one reason why there is so much dissatisfaction with law, but it would appear that the engineer should be the first to reject tradition when it is self-evident that tradition is wrong in the light of present-day development.

The author believes he is safe in saying that there is no best manufacturing method and no best manufacturing tool which is today the same as it was fifteen years ago; that there are very few which are the best today that are the same now as they were five years ago; and further that any manufacturing concern manufacturing any product today exactly the same as it did six months ago, is not pursuing the best possible way, and that if any product has not been entirely redesigned during the last five years, probably the concern manufacturing it is rapidly losing ground.

Savings in Cost Due to Replacing Castings with Arc-Welded Steel Parts

In spite of the tendency to go slowly with changes as radical as that of replacing castings and rivets with welding, there has nevertheless been enormous progress already in its application. At the present time most of the bases of motor-generator sets and motor-driven pumps, and work of a similar simple nature formerly made of cast iron have been redesigned in structural steel. Some buildings have been erected this way, and some bridges have also been made of arc-welded steel. In these cases the saving has ranged from 85 to 15 per cent of the previous cast-iron or riveted-steel cost.

Other examples of this same fundamental scheme of applying arc-welded steel in place of castings are as follows:

In the case of the rotor spider for an electric motor (Fig. 1):

Cast Iron		Arc-Welded Steel	
Cost of iron casting	$6.75	Cost of rolled steel	$1.28
Labor cost	2.94	Labor cost	4.37
Total	$9.69	Total	$5.65
Weight, 70¼ lb.		Weight, 43¼ lb.	

Be sure to bring this copy with you to the Spring Meeting

Another case is that of the end ring for a motor which heretofore has always been made of cast iron. The comparative costs follow:

Cast Iron		Arc-Welded Steel	
Cost of rough casting	$2.89	Cost of rolled steel	$1.126
Labor cost	0.195	Labor cost	0.407
Total	$3.085	Total	$1.533

Another case is a motor-base rail (Fig. 2) which heretofore has always been made of cast iron, the comparative costs being as follows:

Cast Iron		Arc-Welded Steel	
Cost of rough casting	$4.992	Cost of rolled steel	$1.536
Labor cost	0.117	Labor cost	0.617
Total	$5.109	Total	$2.153
Weight, 77 lb.		Weight, 66 lb.	

Another interesting application is that to the motor extension for an elevator base (Fig. 3) heretofore always made of cast iron. The comparative costs are:

	Weight	Cost
Cast iron	184 lb.	$11.04
Arc-welded steel	109 lb.	5.45

A cast-iron wheel for a shop truck has been replaced by one made of arc-welded steel (Fig. 4). In this case a section of T-stock is bent into a circle and welded, a piece of pipe being used for the hub, and the flat stock for the spokes. The comparative cost follows:

Arc-Welded Steel		Cast Iron	
Cost of rolled steel	$0.35	Cost of rough casting	$1.75
Labor cost	0.33	Labor cost	0.10
Total	$0.68	Total	$1.85
Weight, 12.5 lb.		Weight, 25 lb.	

The comparative costs of a number of other parts which have a more or less wide application (since they are indicative of what can be done in the redesign of any part from cast iron to steel) are given below.

Cast Iron		Steel	
12-in. Pulley (Fig. 5)			
Material	$7.90	Material	$1.20
Labor	2.99	Labor	1.85
Total	$10.89	Total	$3.05
Weight, 79 lb.		Weight, 40 lb.	

	Weight	Cost
Wall Box (Fig. 6)		
Cast iron	35 lb.	$2.10
Steel (material $0.42; $0.36)	14 lb.	0.72
Support (Fig. 7)		
Cast Iron	27 lb.	$2.27
Steel (material, $0.42; labor, $0.23)	14 lb.	0.65

It will be noted that in no case is the overhead included in these costs. The reason this is not done is not because of the fact that it will change the final result, but because overhead is a widely varying quantity and in amount is to a considerable extent controlled by the amount of inventory, as inventory, depreciation, storage, and interest charges are invariably a part of the overhead.

It is self-evident that in general the labor cost is higher in the case of arc-welded steel replacing cast iron than it is for the original casting. Therefore, if overhead were included with its usual percentage, the ratio would be more favorable to cast iron, although the arc-welded steel would still be very much cheaper. However, arc-welded steel has one very great advantage over cast iron, and that is, that the inventory cost necessary for a given production will be enormously decreased. Therefore the item of inventory will be decreased by this substitution. Also, the items of patterns, pattern storage, and repair will be entirely eliminated, which also will very greatly affect inventory.

In general, it is safe to say that the items which would reduce overhead would be largely balanced by the increase necessary because of the increased labor cost, so that the total actual figure in both cases, including overhead, would probably, as an average, balance. This is an estimate only, but it indicates what the company with which the author is identified has found to be true.

Much work must still be done in getting the best shapes and the best method of applying these shapes to the work in hand. In the case of buildings there is still much to be done in the design of the joint for most efficient application of the new process, and also in the methods of erection, which become a very serious problem in arriving at the most economical one possible.

Arc Welding Applied to Pipe Manufacture

Another application which is just beginning is the manufacture of pipe by this new process. The advantages here lie in the fact that both the diameter and wall thickness can have any dimensions that are required, and that the material used can have a very wide range of specification without the cost being probably more than two-thirds that of tube-mill or riveted pipe. Because of this, pipe lines in the future will undoubtedly be welded in the making of each section itself as well as the sections welded together in the field.

At the present time a number of arc-welded water lines have been completed, some of them running up close to 100 miles in length, and having diameters ranging up to 6 ft. Yet it is probably safe to say that the average engineer would still put in riveted or lock-bar pipe because that is what has always been put in, in spite of the fact that the new welded-joint procedure will save much money and make a very much more efficient line.

Inconsistency of Permitting Arc Welding in Superheaters or Steam Piping But Not in Boilers

This attitude of mind is probably best illustrated by the present practice in connection with high-pressure superheated steam. The steam must be generated in boilers which are riveted (because state laws permit no other process than riveting being used in a boiler shell), however, the superheater cannot be made tight by rivets because of the very much higher temperature at which it is operated. Also, the steam piping running from the superheater to the turbines cannot be made tight except by arc welding. Therefore we have the rather anomalous situation of the boiler generating the steam being riveted, and the superheater and piping operating at exactly the same pressure but at a higher temperature, being welded.

Another illustration of the same rather unique point of view is shown in the common locomotive. The Interstate Commerce Commission allows very wide latitude in the use of arc welding on locomotives, and there are few locomotives in existence at the present time on which there has not been a very considerable amount of arc welding done both in its manufacture and repair. As is well known, locomotives carry 200 lb. steam pressure, are hand fired, and are racked over the rails at sixty miles per hour. Yet under these conditions very few cases of weld failure have ever occurred, and arc welding has, in the opinion of many railroad companies, increased enormously the amount of time that a locomotive can be kept outside of the repair shop. Yet if one of these welded locomotive boilers is removed from the rails, put on a brick foundation, automatic-stoker-fed, and run at 16 lb. pressure, its owner will be put in jail.

If an arc-welded boiler should explode tomorrow morning and kill a man, it is safe to say that the fact would be broadcast from one end of the country to the other by a number of journals. The fact, however, that on an average there is one man killed every day by the explosion of a riveted boiler, does not even gain passing notice. This is not cited as an argument in favor of welding, but merely as a fact.

Practically without exception every automobile depends for its success on electric arc welding, and if the electric welds failed in service most automobiles would be wrecked, with enormous loss of life. In fact, it is safe to say that practically no automobile (with the exception of the Ford) is in operation today which does not have such parts of it arc welded that if a weld failed a serious accident would result.

Many automobiles not only have the rear-axle housing completely made by arc welding, but in almost all cases also the driveshaft and brake hangers, thus the ability to start, run, stop, and stay on their wheels all depends on arc welding.

It is safe to say that if arc welding in the automobile industry were found defective, it would be ten times more destructive to life than if all the boilers in the country were to explode at once.

The author is citing these cases in the hope that engineers may be led to take a logical view of this great question. It is certainly true that if arc welding is dangerous it ought to be eliminated in any place where its failure will endanger life, but it certainly is a very poor recommendation for engineers and the engineering profession to allow conditions to exist such as are existing at the present time in which arc welding is eliminated from a boiler shell but used in the superheater of the same boiler; where arc welding is eliminated from air receivers but is used completely in a cracking still; where arc welding is satisfactory on a locomotive boiler but unsatisfactory and forbidden by law in a stationary boiler; where arc welding is prohibited by law from being used in a tank car but specified in the tank into which the tank car is emptied.

The engineering profession is founded fundamentally on logic; if an engineer is not logical, to whom, then, shall we turn?

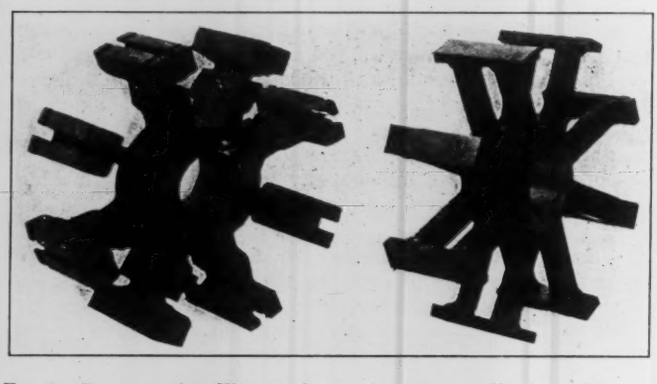

FIG. 1 CAST AND ARC-WELDED ROTOR SPIDERS FOR ELECTRIC MOTORS

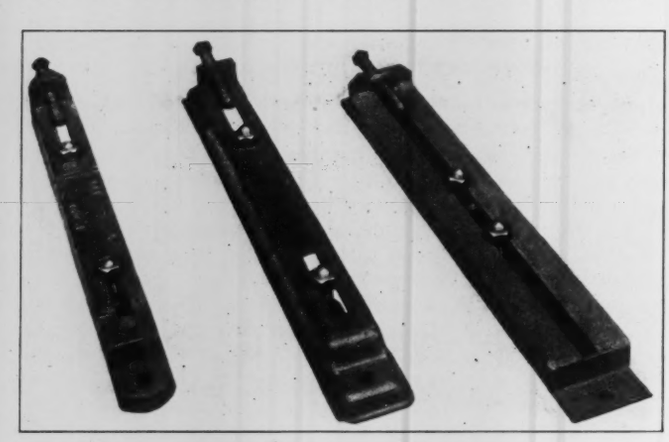

FIG. 2 CAST AND ARC-WELDED MOTOR-BASE RAILS

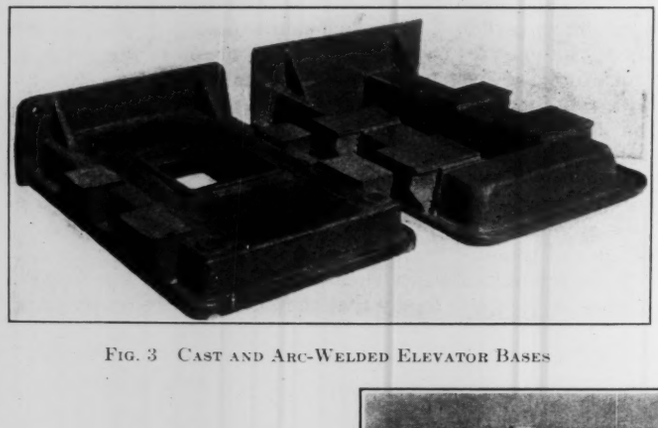

FIG. 3 CAST AND ARC-WELDED ELEVATOR BASES

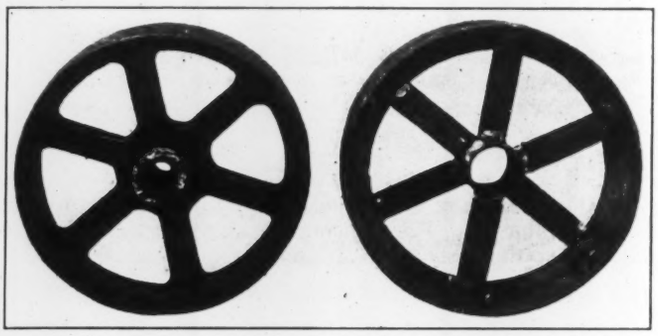

FIG. 4 CAST AND ARC-WELDED WHEELS FOR SHOP TRUCK

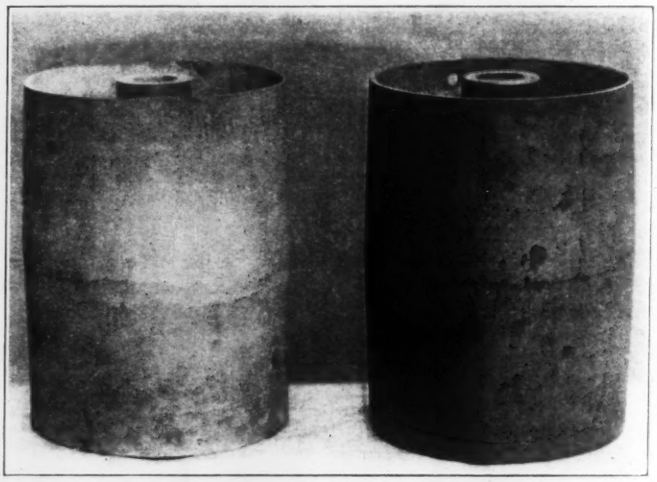

FIG. 5 12-IN. PULLEYS, ARC-WELDED AND CAST

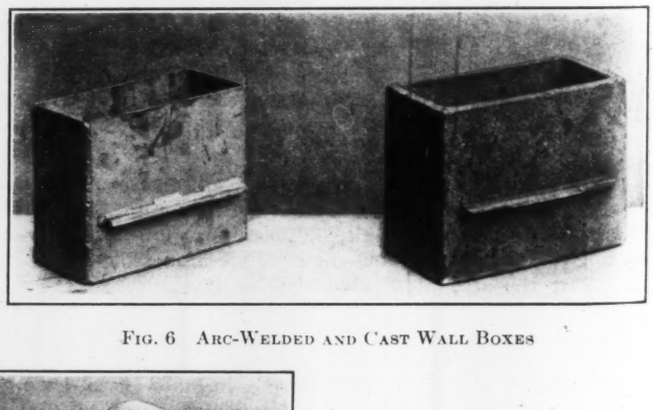

FIG. 6 ARC-WELDED AND CAST WALL BOXES

FIG. 7 ARC-WELDED AND CAST SUPPORTS

Hysteresis Relative to the Operation of Mechanical Springs

By JOSEPH KAYE WOOD,[1] NEW YORK, N. Y.

Mechanically, hysteresis, or the lag of observed effects behind their causes, has not been considered as extensively as in electrical design. Springs, and spring systems in particular, are shown by the author to be associated with physical hysteresis of three characteristic types. Thus mechanical, hypo-elastic, and hyper-elastic hysteresis are described, and the former is shown to be important from the standpoint of automotive riding quality. Mechanical hysteresis is caused by external agencies, while both hypo-elastic and hyper-elastic hysteresis are caused by the internal behavior of crystalline structures. Hypo-elastic hysteresis is due to an internal friction of the solid type, presumably at the grain boundaries, while hyper-elastic hysteresis is due to slip or plastic flow and hence has a characteristic time factor. The concept of hypo-elastic and hyper-elastic energies with their corresponding hysteresis effects enables a better understanding of certain physical properties of metals, particularly those important to mechanical springs. Although elastic-hysteresis effects are of a much lower order than mechanical-hysteresis effects, they are sufficiently large to be given serious consideration in the design of measuring devices.

IN 1881 Ewing[2] announced to a gathering of the Royal Society of London his selection of the word "hysteresis"—taken from the Greek and meaning "to lag behind"—to designate a type of phenomena characterized by a noticeable lag of observed effects behind their causes. A study of the behavior of engineering materials will show that in general, magnetic, thermoelectric, strain, and other similar effects vary with their corresponding causes in accordance with certain definite laws, modified in each case by hysteresis. Philosophically, every law of cause and effect must stand modified by hysteresis due to the element of time; that is to say, cause and effect cannot take place simultaneously, there must be a finite interval of time between the two. The relative magnitude of this period of time is the determining factor as to the practical significance of the variation of a law due to hysteresis. With magnetic hysteresis, which is the lag of magnetization behind magnetizing force, the variation is sufficiently large to warrant a practical consideration of this effect in electrical design. Although mechanical engineers of the past have not been concerned with hysteresis effects enough to warrant their practical consideration, recent developments and refinements in mechanical design, particularly in regard to springs, have made it necessary to take into account certain variations due to physical hysteresis. Consequently any existing publications on this relatively new subject are, with a few exceptions, of a purely academic character, although in most cases the studies, whether pursued through mathematical or experimental channels, have yielded highly important results. But, as is often the case where the practical demand is lacking, these results have served no widespread useful purpose except to be chronicled in scientific proceedings for future use. The author, believing that a practical demand for knowledge on physical hysteresis is now at hand, has endeavored in the present paper to cover all phases of this study in a logically coördinated and systematic arrangement without a display of the mathematics, particular attention being given to its relation to the operation of mechanical springs. Much of the work is new, but use is made of whatever knowledge is available to support certain vital assumptions. It is hoped, therefore, that the publication of this paper will bring about, first, a clear-cut status and subsequent wider recognition of the subject of physical hysteresis; second, through considerable discussion a standardization of definitions and nomenclature, also a clear and systematic understanding of all fundamental principles involved; and third, the promotion of research on physical hysteresis by both private and national efforts on a coöperative basis.

Physical hysteresis is of three kinds, namely, mechanical hysteresis produced by any external interference, such as solid or fluid friction, to a given law of cause and effect; hypo-elastic hysteresis, resulting from internal friction of the solid type; and hyper-elastic hysteresis, resulting from internal friction of the plastic or viscous type. Mechanical springs used in sensitive instruments or machines for measuring weight, length, pressure, electrical quantities, etc., under present-day requirements must be carefully designed so as to eliminate as far as possible all noticeable trace of mechanical and elastic hysteresis. Vehicle suspensions for automotive or railway purposes and other types of shock-absorbing devices must, on the other hand, be designed with a large mechanical hysteresis in order to assure absorption of oscillatory energy, which requirement is essential to the attainment of good riding quality.

In the study of hysteresis, promotive causes are expressed as forces either directly or in terms of reactive forces per unit of area: that is, in stress units; while observed effects are expressed as movements which may be simple displacements, deformations, or deformations per unit of length: that is, in strain units. Hysteresis may therefore be looked upon as a lag in movement or strain caused by a change in force or stress. Consider Fig. 1, which describes diagrammatically the fundamental meaning of hysteresis and also of the hysteresis loop. The straight line $OhnO'$ represents a given law of cause and effect without hysteresis, although in the absolute sense such a condition cannot exist, as will be demonstrated later. Assume, now, that under a certain condition of speed and a given scale of plotting actual observations lie along the line OgO' for ascending magnitudes of force and along the line $O'mO$ for descending magnitudes of force. Although the latter line represents observations of descending magnitudes of movement for descending magnitudes of force, the directions of both increasing cause and effect are reversed, because it is the magnitudes of *change* in force and corresponding magnitudes of *change* in effect that determine the directions of cause and effect. Take, for instance, the force e which intersects the direct ascending line OgO' at g, and the "hysteresisless" line $OhnO'$ at h; the change in magnitude of force is equal to eO. Without hysteresis the corresponding change in magnitude of movement would be equal to sO, but since there is hysteresis, the change observed is somewhat less, being equal to rO. The lag sr in movement is a measure of hysteresis for the change of force so noted. Similarly for the force e' which intersects the direct descending line $O'mO$ at m and the "hysteresisless" line $OhnO'$ at n, the change in force is equal to $e'O$. The corresponding hysteresis is equal to $s'r'$. Assuming Oe equal to $O'e'$ and Or equal to $O'r'$, we find that in the closed hysteresis loop $OgO'mO$ the "hysteresisless" line $OhnO'$ divides that loop in halves which are symmetrical only when one of the halves is rotated in the vertical plane of $OhnO'$ through an angle of 180 deg. and the O'-point of one coincides with the O-point of the other. This characteristic of the hysteresis loop will later play an important part in the explanation of the general shape of certain types of loops.

The hysteresis loop in Fig. 1 represents a complete cycle of movements from zero to maximum and back to zero again, due to a complete cycle of forces varying from zero to maximum and back to zero again. Since the forces increase and decrease between zero and a finite positive value, the loop may be called a "positive hysteresis loop" with the line OgO' labeled as the $+A$ or positive ascending curve, and the line $O'mO$ as the $+D$ or positive descending curve. Lag sr would then be called a $+A$ hysteresis, and lag $s'r'$ a $+D$ hysteresis. In an actual determination of the hysteresis loop, however, the "hysteresisless" line $OhnO'$ would not be obtained, hence any hysteresis value would be the sum of the $+A$ and the $+D$ values in the same horizontal force line such as, for example, xy. The latter or total hysteresis for any given force

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[2] J. A. Ewing, Proceedings of the Royal Society of London, no. 216, 1881, p. 22.

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in a positive hysteresis loop would be known then as a + or positive hysteresis. The maximum horizontal width of the loop is the maximum hysteresis, and the average width of the loop is the average hysteresis. There is some question in the author's mind as to whether the maximum hysteresis should not be made up of the sum of the maximum $+A$ and maximum $+D$ hysteresis irrespective of whether they are in the same horizontal line, since the same magnitude of cause or change of force is involved in each of these components.

Suppose now the force is also made to vary between zero and a finite negative value (opposite direction). The "negative hysteresis loop" identical with the first or positive hysteresis loop would be obtained in which the various components would be named as indicated in Fig. 1.

What causes hysteresis? The answer to this question requires the consideration of two principal factors, namely, the element of

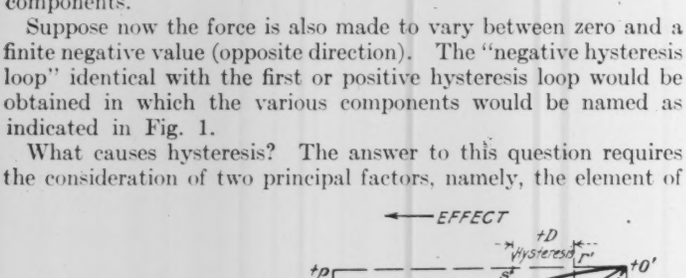

Fig. 1 Diagram Explanatory of Hysteresis and the Hysteresis Loop

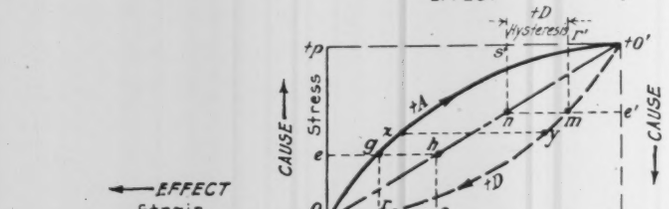

Fig. 2 Hysteresis Always Present in the Absolute Sense

time and a superimposed series of minor force cycles. In Fig. 2(*a*) is shown a "hysteresisless" relation of cause and effect OO' upon which we may plot any minor relation or cycles of minor relations that we suspect of being the cause of hysteresis. Beginning with Fig. 2 (*b*), which shows the probable effect of infinitesimal time lag plotted to an extremely large scale, we shall consider the previous statement that "cause and effect cannot take place simultaneously." If this statement be true and we take an increment of force $O1$ small enough, then the relation between force and movement must follow along the path $O12$ instead of along the path $O2$, and so on with the remaining increments of force both for ascending and descending values. Due to the lack of perfect homogeneity in the general order of things of practical dimensions, we should expect to have many of these stepped paths mounted on the line OO' with an infinite variety of phase angles with respect to each other. This would give rise to the hysteresis loop $O140'560$ of extremely small maximum width—indeed so small in most cases as to be incomprehensible. The practical determination of this relation would yield naught but a straight line, and hence no visible hysteresis. Nevertheless hysteresis must be present in all transitions from cause to effect.

Time effects of sensible proportions such as are associated with viscosity produce visible hysteresis loops of shapes somewhat different from the "philosophical" hysteresis loop in Fig. 2 (*b*), as will be seen in the discussion to follow. Assume in Fig. 3 the basic relation OO' and a sensible range of speed and scale of plotting of the applied force. Now suppose that at the end of time t the force f produces a movement d, and then, without permitting any lapse of time for the completion of the full movement d_1 an additional increment of force f is applied. By applying the force in this manner at the uniform rate of f/t, points O, 1, 2, 3, etc., are obtained, and by assuming different values, say $f/t_1, f/t_2, f/t_3 \ldots f/t_n$, an infinite number of different-sized hysteresis loops may be obtained ranging from a maximum of $Of_pO'd_pO$ to the minimum or zero loop of $OO'O$. It is assumed in all of the above cases that sufficient time is allowed for the completion of the movements hO' and $h'O$.

Next assume that for a uniform increase of force with respect to time, the speed of movement is proportional to the magnitude of force. This is the law of viscosity, which may be expressed symbolically as follows:

$$aF = \frac{M}{t} \qquad [1]$$

where F is the force, M the movement, t the time of movement, and a a constant. Consider Fig. 4 (*a*) where the force F is applied to the piston p acting in the cylinder c filled with a liquid. So long as the constant force F is applied uniformly the movement of the piston will be uniform. But let the spring S_1 in Fig. 4 (*b*) be

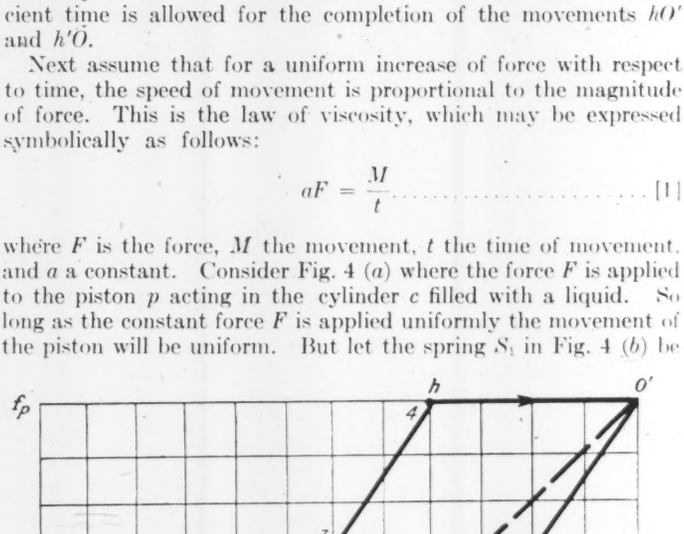

Fig. 3 Simplest Hysteresis Depending upon Time

deflected against the piston at a constant rate. Then as the force increases the rate of movement also increases, giving rise to the type of hysteresis loop shown in Fig. 4 (*c*). Both the A and D lines of this loop are logarithmic curves obeying the expression [1], while line OO' is the force-movement relation of the spring S_1, assuming that the spring itself has no appreciable elastic hysteresis. Movements for equal force increments in this case increase in geometric progression as compared with the case in Fig. 3, where they increase in arithmetical progression. The loop in Fig. 4 (*c*) is called a mechanical-hysteresis loop of the viscosity type.

Considering now the other principal factors in the development of hysteresis, namely, superimposed minor force cycles, let us refer to Fig. 5 (*a*), which represents a type of loop produced by a condition analogous to the arrangement shown in Fig. 5 (*b*). The resistance offered by the large spring S_1 corresponds to the basic relation OO', while the force of friction acting at the surfaces

f' due to the normal pressure exerted by the springs S_2 represents the constant force f_h. The latter force superimposed on the basic relation OO' gives the $f_h c$ line, and on descending the $cO'O$ line is obtained. Suppose now that the core E in Fig. 5 (*a*) is free to move in the rigid cylinder W until the shoulder m bears against the end of the cylinder. Movement b acts as "backlash" in so far as the force of friction is concerned, but not so in the case of the basic force supplied in this case by spring S_1. The effect of this condition alone would be to flatten the ends of the loops to straight lines, but if a series of these arrangements acting out of phase with each other be considered, then the ends of the loop will taper gradually to points as indicated. The latter modification occurs in an aggregate of crystalline metal grains, although this involves hysteresis of an elastic or internal type which will be taken up later.

The type of loop described might be termed a frictional hysteresis loop, and if the basic relation and friction are external to each other—that is to say, involve two separate and distinct bodies such as our example in Fig. 5 (*b*) happens to be—then we have mechanical hysteresis. On the other hand, if the basic relation and friction of the solid type are inherent properties of the same bodies or material, then we have hypo-elastic hysteresis, while if the basic relation and frictional interference of the viscous or plastic type are inherent properties of the same material, then we have hyperelastic hysteresis.

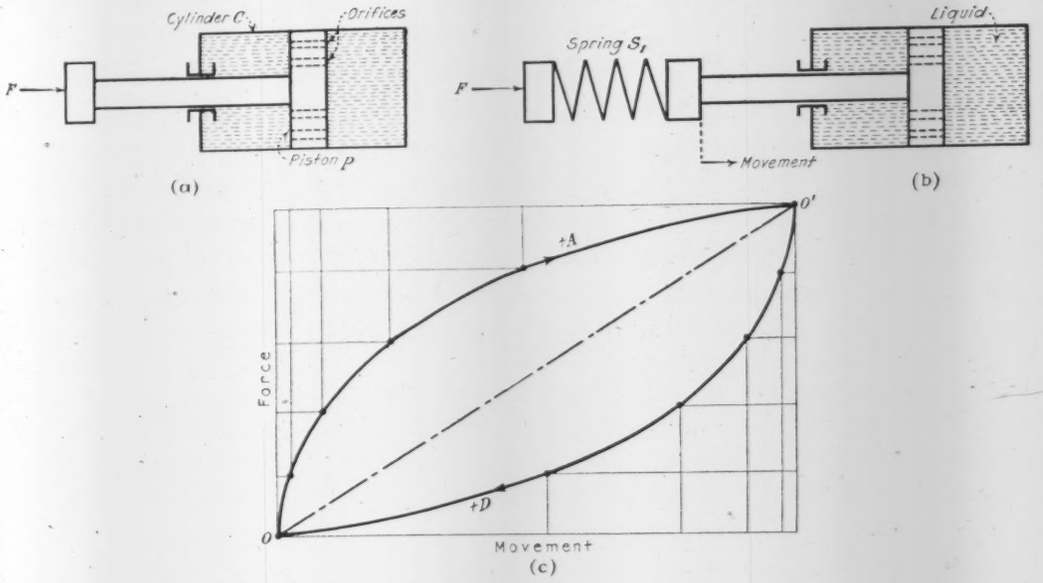

FIG. 4 MECHANICAL-HYSTERESIS LOOP OF THE VISCOSITY TYPE

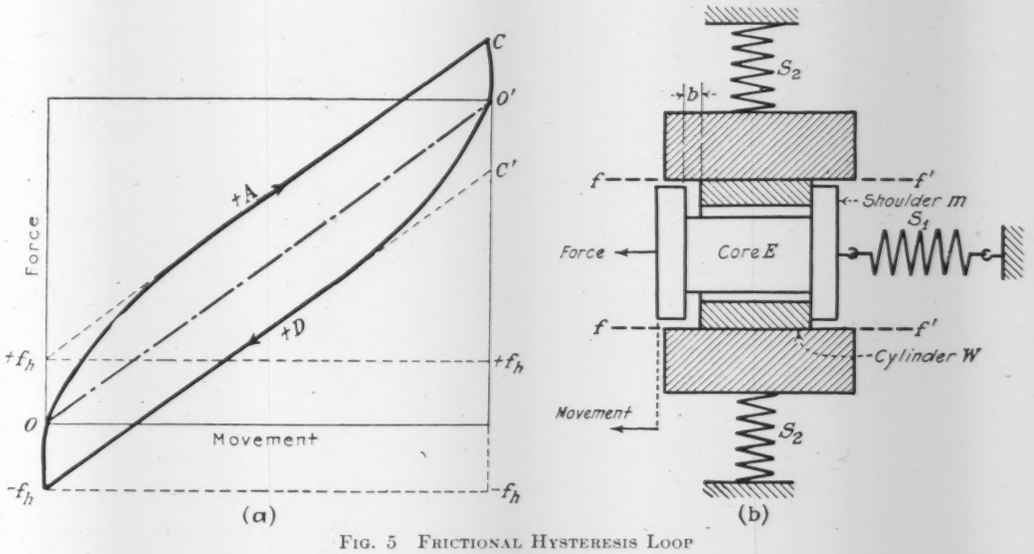

FIG. 5 FRICTIONAL HYSTERESIS LOOP

MECHANICAL-SPRING CHARACTERISTICS

Springs as a definite class of mechanical elements were first recognized by Robert Hooke of England about 250 years ago, at which time he formally announced to the King of England his discovery of a new theory of springs and gave a practical demonstration of it by presenting him with the first watch ever driven and controlled by springs. Since that time springs have been adopted for an enormous variety of uses, although the existence of an art in connection with mechanical springs in general has been rather obscure. Nevertheless springs are thoroughly woven throughout all industries, and it would be impossible to estimate their importance to the successful operation of all types of machines, engines, instruments, etc., or their effect on human welfare and safety. An enormous number of springs are manufactured each year by both spring manufacturers and users, but for such a commonly used mechanical element, engineers in general appear to know very little about them. The most obvious distinction between a spring and other types of mechanical elements is the relatively large retroactive (elastic) deflection for a given stress. In the development of certain spring formulas this fundamentally basic distinction has been lost sight of; that is to say, many of the formulas have simply been borrowed from the practice of rigid mechanical design, in which the fundamental assumptions specifically point out that the deflections for all stresses must be of a relatively small magnitude, a requirement contrary to the very nature of springs. Should we be absolutely certain that no new or unusual factors are introduced into the performance of springs due to the relatively large deflection, even though the stress be of the same magnitude as in the rigid member whose corresponding deflection is extremely small? One case in particular where relatively large deflections have a profound influence is in the development of hysteresis loops, either of the mechanical, hyper-elastic or hypo-elastic types. Hysteresis, as has been stated, is the lag of deflection given in length units, hence if the total deflection be extremely small, the maximum hysteresis would be too inappreciable for practical consideration. Furthermore the accuracy of deflectional control of springs often has a very important function, as, for example, in the measurement of contact pressures for length-measuring machines. It is obvious, therefore, that since deflection is the outstanding component of both spring action and hysteresis, these two manifestations have an important mutual relation.

MECHANICAL HYSTERESIS

Mechanical hysteresis is the lag in movement following the application or release of force to a mechanism whereby the interference causing the lag is external to the moving part. Friction between pins or pivots and their respective bearings is one source of mechanical hysteresis because the bearings are external to the pins or pivots. Hysteresis of this source can be greatly minimized by proper design of the pivots or pins and by lubrication. In cases where very large hysteresis effects are desired, one must turn to

sources of the mechanical type, because at best hysteresis of the elastic type is comparatively small. It is only in cases where hysteresis must be eliminated to a high degree, such as in measuring instruments, that we are interested in the characteristics of hypo-elastic or hyper-elastic hysteresis. Of course a study of the latter has an important bearing on problems relating to the fatigue of metals or the theory of vibrations, whether of small amplitude or in connection with the oscillations of springs.

Therefore, when a large measure of hysteresis is desired, it is possible to design a mechanical-hysteresis loop of a size and shape best suited for the purpose. Consider, for example, the spring suspension of an automobile. Leaf springs, consisting of a number of long, narrow steel strips placed one on top of the other and held together by a band and several clips, are interposed between each wheel and the frame of a vehicle. The frame and all other masses in contact with the frame except the springs constitute the "sprung" weight, while the remainder of the vehicle, such as the wheels and axles constitute the "unsprung" weight. Now when the vehicle passes over an obstruction or rolls into a rut on the road, the sprung weight receives an upward or downward acceleration, depending upon the speed at which the vehicle is progressing. This starts a vertical harmonic oscillation of the sprung weight about its static position, i.e., the position when the vehicle is not in motion. Under the impossible theoretical assumption of no hysteresis in the system, the oscillation would be sustained indefinitely at the original amplitude provided no further obstructions or ruts were to be encountered. Fig. 6 shows the graph that would be obtained under the previous assumption if the vehicle maintained a constant speed on a perfectly smooth road after encountering the obstruction and a projecting pencil attached firmly to the sprung weight were made to bear against a vertical plane parallel to the road. The sustained oscillation would obviously be unbearable for the passengers, even for a short period, and would also decrease the life of the car considerably. As a matter of fact, however, we know that these oscillations are actually destroyed in a very short time, but in order to understand this "damping" action, particularly in its relation to hysteresis, let us assume that the job of designing and producing this damping action is assigned to an engineer.

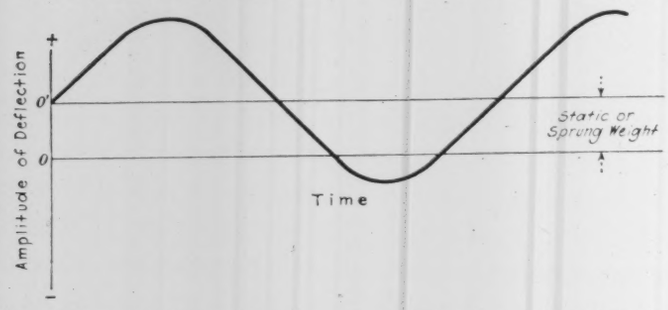

FIG. 6 GRAPH OF AUTOMOBILE OSCILLATION AFTER STRIKING AN OBSTACLE

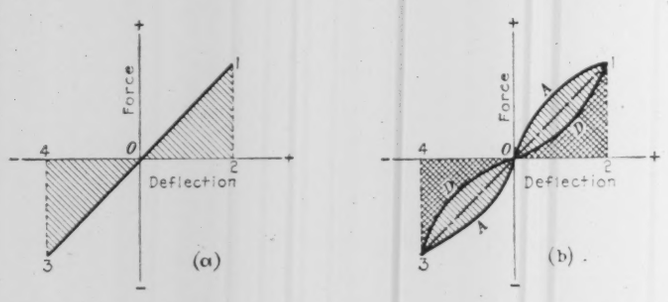

FIG. 7 INTERCHANGING ELASTIC ENERGIES OF AN OSCILLATING SPRING

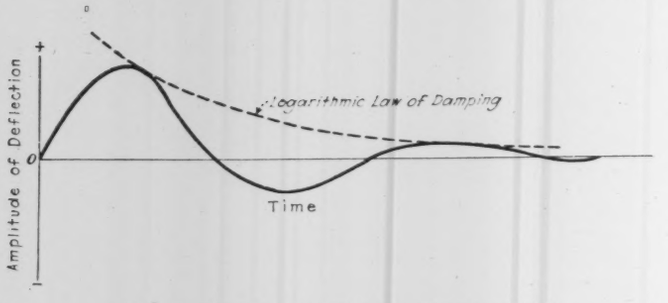

FIG. 8 METHOD OF DAMPING OSCILLATIONS

Without hysteresis of any kind, including that due to the resistance of the air, a spring once set in oscillation will keep on so vibrating indefinitely at a constant amplitude. This should be the case, because upon release of the spring at 1 in Fig. 7 (*a*) the stored or potential energy *0120* is gradually changed to kinetic energy, until at *0* it is entirely changed into an equal amount of kinetic energy. The latter energy is then expended and changed into an equal amount of stored or potential energy *0340* which deflects the spring in the opposite direction. So long as the exchange of these two equal amounts of potential energy takes place between the positive and negative positions without any loss, the oscillation once started should keep on indefinitely at the constant amplitude 12 or 34. The reason for this perpetual flow of potential energy back and forth between the positive and negative positions through the medium of kinetic energy is because of the assumption that the action of the spring is 100 per cent retroactive and the hysteresis is zero. The question arises, how can the motion just described be stopped? A logical method would be simply to introduce a non-retroactive force so as to oppose the spring's motion at all times. Upon loading from *0* to 1 in Fig. 7 (*b*) the total energy expended would be *OA120*, the area *OA10* being due to the added non-retroactive force. Upon unloading from 1 to *0*, the total retroactive energy received back would be *OD120* because the area *01D0* represents the added retroactive force acting in the reverse direction so as to oppose the complete recovery of the retroactive energy *0120*. Therefore of the total energy *OA120* put into the system, only *OD120* is retroactive or recoverable. If this energy were available for full conversion into potential energy in the opposite direction, area *0340* would be less than area *0120* but equal to area *OD120*. Thus amplitude 12 is reduced to amplitude 34 for half an oscillation. Now assume that the non-retroactive force is made to oppose the spring in the negative direction also, thus reducing the delivered energy *OA340*, equal in this case to area *OD120*, to the retroactive or recoverable energy *OD340*. This cutting down of the recoverable energy at each half-oscillation due to the losses represented by the hysteresis loops gradually reduces the amplitude to zero, thus destroying the oscillations of the spring. The speed at which these oscillations are damped or destroyed due to hysteresis depends upon the nature and relative magnitude of the non-retroactive force introduced for this purpose. The manner in which these oscillations are damped is shown by the graph in Fig. 8. Under actual conditions the free oscillations of a spring are always destroyed in a relatively short time by hypo-elastic hysteresis and, if the initial stress is high, by hyper-elastic hysteresis. In certain exceptional cases air resistance may be sufficiently great to constitute an external or mechanical hysteresis. For vehicle suspensions this natural or internal hysteresis of the spring is insufficient to damp the oscillations quickly enough to insure comfort.

In the case of leaf springs, however, it very fortunately happens that considerable friction exists between the leaf surfaces, which increases somewhat as the load upon the spring increases. This forms a mechanical-hysteresis loop of greater magnitude than the inherent elastic-hysteresis loop mentioned previously, and accordingly is more effective in destroying oscillations that are so uncomfortable to passengers. Since this friction is uncontrollable and variable due to improper shaping of the leaves, variation in pressure exerted by clips and bands, and frictional condition of surfaces, suspensions for modern high-speed vehicles, i.e., automobiles and railway cars, have in general been very unsatisfactory from the standpoint of riding comfort. In the automotive industry this condition has given rise to a distinct class of accessories that might be given the general name of oscillatory controllers and which in themselves represent a business of many millions of dollars. This class of accessories, comprising as it does devices under such names as snubbers, shock absorbers, stabilizers, checks, etc., are merely means to introduce less variable and more controllable hysteresis sources than the natural sources of friction exerted between the leaves. In one case the device consists of an ad-

justable clamp that squeezes the leaves together so as to increase and control the interleaf friction; but this method is not to be encouraged, because it simply aggravates a serious source of our enormous spring breakages, that is, "ripple" bending. The general purpose of all "oscillatory controllers" is to control and increase the mechanical hysteresis of an oscillatory spring system, although very few of such devices on the market have apparently been designed with a clear understanding or application of this principle. The mediums for supplying the various types of non-retroactive interferences to the motion of the oscillating spring are confined air or liquids, rubber, and solid friction, augmented in some cases by auxiliary spring action. These additional sources

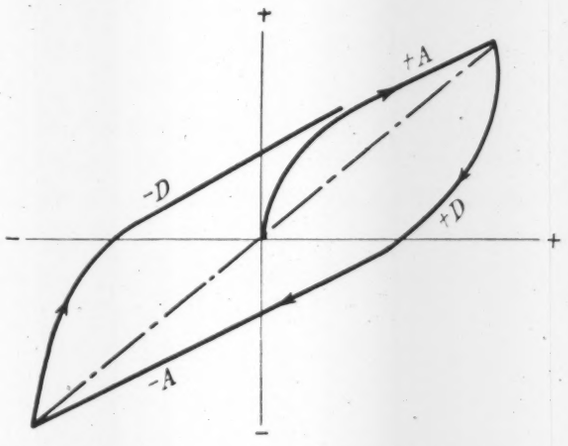

FIG. 9 FULL HYSTERESIS LOOP OF THE VISCOSITY TYPE FOR A COMPLETE OSCILLATION

of mechanical hysteresis produce hysteresis loops of considerable magnitude and of a variety of shapes, depending upon the medium or combination of mediums employed. Viscous mediums produce loops of the types shown in Fig. 4, and solid-friction mediums of the kind shown in Fig. 5, while a combination of both mediums produces a loop having the combined characteristics of both Fig. 4 and Fig. 5. Mechanical-hysteresis loops of the viscous type are most effective in quickly destroying oscillations, because the size of the hysteresis loop automatically adjusts itself to the periodicity of the oscillation. In the viscosity loop shown in Fig. 4, the curve $+A$ rises more steeply as the speed of force application (period of oscillation) is increased, which likewise increases the area of the loop. If the mechanical hysteresis of a suspension is due mainly to a viscous loop of the type shown in Fig. 4, then the amplitudes of succeeding oscillations will decrease along the logarithmic curve shown in Fig. 8, which corresponds to the logarithmic curve $+A$ and $+D$ of Fig. 4, a characteristic common to viscosity loops. Fig. 9 shows the full hysteresis loop of the viscosity type for a complete oscillation.

In predetermining the magnitude and shape of the hysteresis loop in an oscillatory controller, one must know the representative initial amplitude and period of the oscillating spring, and the rate of damping necessary for the physiological and psychological satisfaction of the average passenger. The controller must be so designed that its force characteristic "meshes" properly with the spring characteristic. Very few if any of the devices on the market have apparently been designed with full consideration being given to these factors.

HYPO-ELASTIC AND HYPER-ELASTIC ENERGY

Elastic hysteresis is the lag in deflection (or strain) following the application or release of a load (or stress) on a body in which the interference causing the lag is an inherent property of the material composing that body.

Elastic hysteresis is termed "hypo-" when the maximum applied stress is equal to or less than the true proportional limit of the material, and "hyper-" when the true proportional limit is exceeded.

The terms "hypo-" and "hyper-" were first applied to the stressing of materials by the author in 1923[3] in an effort to give a clear picture of the mechanism of overstrain and to describe the characteristic shape of the general stress-strain diagram. These terms have the following significance: Fig. 10 represents a metallic crystalline grain in the ideal isolated cubic form and of unit dimensions. In (*a*) the grain is deformed elastically by the shearing couple P_E and in (*b*) slip has occurred on the weakest crystallographic plane due to a very slight increase of force beyond P_E. Deformations d_{E1} and d_{E2} are elastic and are practically equal to d_E, while d_p is plastic. The total deformation d_t is therefore equal to $d_p + d_E$, and the stress P_E is the true proportional limit. Now according to the law of plastic (viscous) flow, the plastic deformation will keep on increasing at a constant rate of speed provided there is no further increase in the force P_E. By the same law, increasing the force to P_T [Fig. 10(*b*)] increases the speed of deformation, but while this is taking place the blocklets m and m_1 are being deformed elastically to an amount d_{E3}. Actually slip on the plane OO' will eventually come to a stop and the increased elastic strain on the blocklets will cause slip on the next weakest plane within either of them, thus repeating the process just described. As the cubic grain is of unit dimensions, Fig. 10 (*d*) gives the stress-strain diagram for the case shown in Fig. 10 (*a*), Fig. 10 (*e*) for Fig. 10 (*b*), and Fig. 10 (*f*) for Fig. 10 (*c*). Referring to Fig. 10 (*e*), the equal

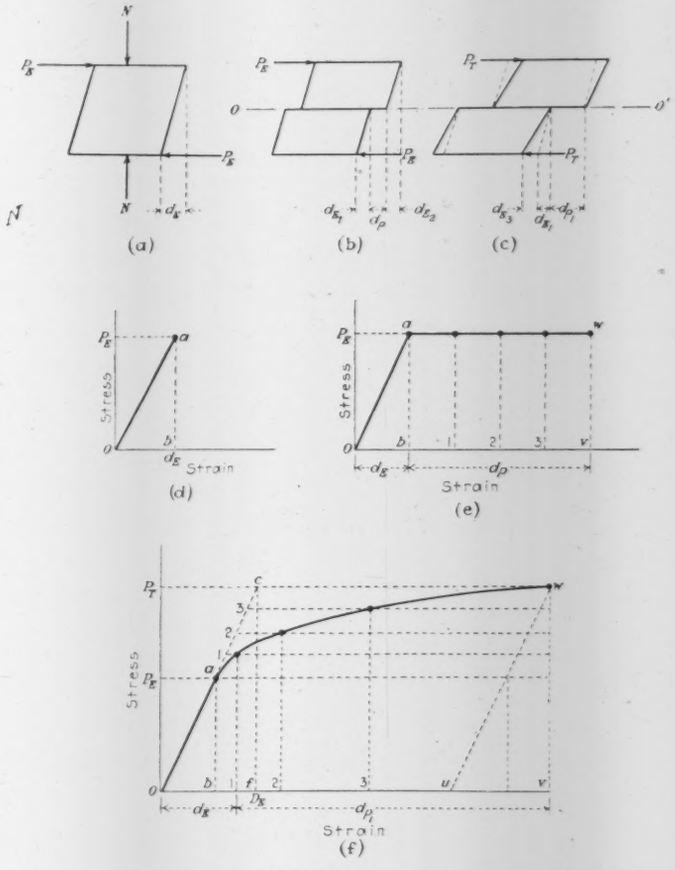

FIG. 10 DEFORMATION OF METALLIC CRYSTALLINE GRAIN UNDER STRESS

deformations $b1$, 12, 23, and $3V$ occur in equal intervals of time so long as P_E is kept constant, but in Fig. 10 (*f*) the uniform increments of force $a1$, 12, 23, and $3c$ cause the logarithmic increase in deformation of $b1$, 12, 23, and $3V$ due to the previously stated fact that in plastic flow the rate of deformation beyond P_E (load of initial flow) varies directly as the increase in force. From this it is evident that elastic action does not stop at P_{E1}, the true proportional limit, but continues right up to the maximum load P_T (tensile strength) simultaneously with plastic action. The action may be viewed in this manner: up to the load P_E the elastic energy *Oab* alone is expended; beyond P_E the additional elastic energy *acfb* is expended in conjunction and simultaneously with the total plastic energy equivalent to area *OawuO*. The elastic energy *Oab* is called hypo-elastic energy, and energy *acfb* is called hyper-elastic energy.

[3] Overstrain in Metals, Joseph Kaye Wood, Trans. A.I.M.E., 1923 pp.

Briefly summarized, hyper-elastic energy generates and accelerates the plastic energy represented by the plastic area in the diagram. First the hypo-elastic energy, is developed, then follows the development of hyper-elastic energy, but while the latter is under development it is also generating and accelerating the growth of plastic energy. Graphically the stress-strain diagram beyond the hypo-elastic area *Oab* may be traced by giving the hyper-elastic area a uniform motion of translation to the right, causing the line *ac* to cut the verticals 1,2,3, and *c*, which points of intersection determine the stress-strain diagram.

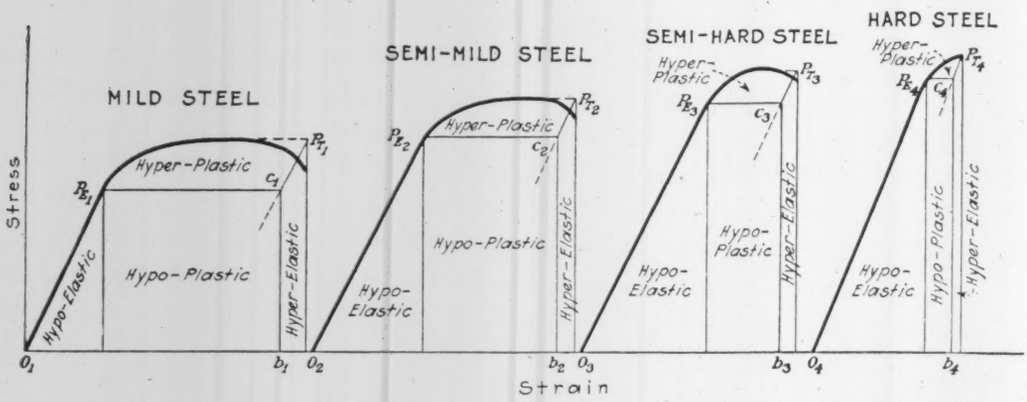

FIG. 11 FOUR TYPICAL STRESS-STRAIN DIAGRAMS OF STEELS OF DIFFERENT CARBON CONTENT

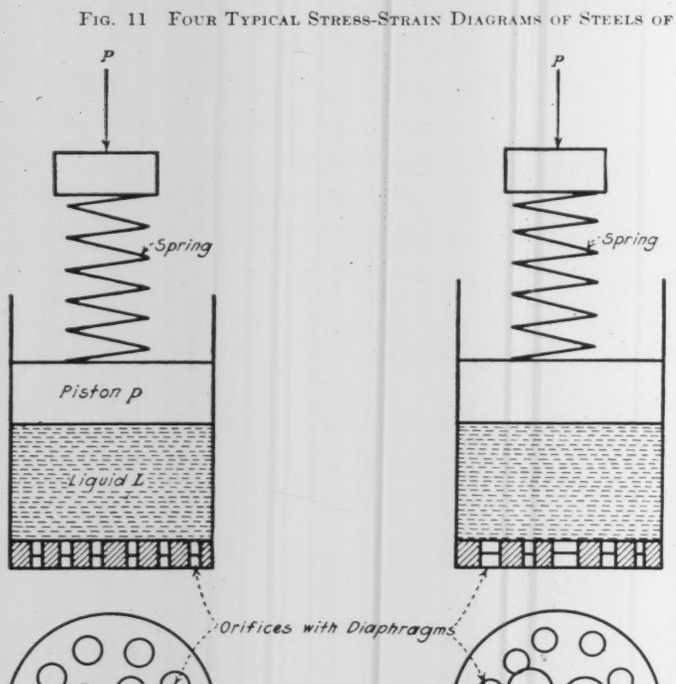

FIG. 12 MODEL ILLUSTRATING THE REASON FOR THE YIELD POINT

In Fig. 11 four typical stress-strain diagrams of steel with different carbon contents are shown in which the usual yield-point dips of the mild and semi-mild steel diagrams are temporarily ignored for the sake of simplicity, since they are not vital to the discussion in hand. Hypo- and hyper-elastic areas are so marked, with the latter area in the final position after the completion of its motion of translation from the initial position adjacent to the hypo-elastic area (*c* coincident with P_E). The total plastic energy is represented by the areas marked hypo- and hyper-plastic, the former being generated by the maximum load P_E (true proportional limit) of the hypo-elastic energy and the latter by the variable load of the hyper-elastic energy (neglecting the small area of the upper right-hand corner of the diagram for mild or semi-mild steel). From these principles it is evident that static strength can only be derived from elastic energy, while plastic energy merely prevents a state of division. Impact strength is derived from elastic strength augmented by the reaction of "choked" plastic action against the normal internal pressure that holds the blocklets of Fig. 10 together. Hence mild steel under impact would break like hard steel under static load, i.e., without elongation, and at a higher equivalent load.

Under actual conditions a piece of metal consists of an aggregate of many crystalline grains oriented as regards their crystallographic planes in all possible directions. Furthermore the aggregate may be a mixture of soft and hard grains, so that in considering all possible variable factors, the net properties of a piece of metal are statistical results of the summation of the properties of individual grains. However, since mild steel has a preponderance of soft grains and had steel a preponderance of hard grains, there should be certain characteristic differences in the net properties and hence the stress-strain diagrams of these two steels. In the mild steel each grain reaches its first slip load, i.e., true proportional limit of the individual grain, at almost the same time, hence the proportion of hyper-elastic energy will be high; while in hard steel the hard grains prevent early slip in the softer grains, particularly if the order of dispersion is high, and hence also give a high ratio of hyper-elastic energy. An even dispersion of both hard and soft grains, i.e., eutectoid steel, would give the lowest ratio of hypo-elastic to hyper-elastic energy.

In pure metals containing nearly all one kind of grains, usually soft, there is a pronounced dip and rise in the diagram following the load P_E which is called the yield point. This may be explained by the fact that since nearly all grains experience slip simultaneously, there is an inertial effect which causes a momentary decrease in stress. The following analogy may serve to illustrate this process. Suppose in Fig. 12 a pressure is brought to bear against the piston through the spring *S*. The bottom of the cylinder has a number of holes closed by diaphragms; in (*a*) the diaphragms all are of equal strength, while in (*b*) they have different strengths. At first, increasing the pressure merely deflects the spring, which corresponds to the growth of hypo-elastic energy. When the pressure reaches a value of P_E the weakest of the diaphragms breaks, allowing the liquid to flow, which corresponds to first slip on the weakest crystallographic plane. Further increase of pressure increases the rate of flow and deflects the spring further, which corresponds to the acceleration of plastic action and to the development of hyper-elastic energy. Now in (*a*), since all of the diaphragms are nearly equal in strength they will break almost simultaneously, which causes the pistons to drop a slight amount suddenly and hence to momentarily decrease the load on the spring, giving what corresponds to the yield-point dip in a stress-strain diagram for mild steel. In (*b*) only a few diaphragms break at first, followed by a more gradual breaking of the remainder, thus preventing the sudden drop of the piston and consequent momentary decrease of the load, and giving what corresponds to the absence of the yield point in a stress-strain diagram for hard steel. Actually ultra-sensitive experiments[4] have shown that the yield-point dip has a slight fluctuation superimposed upon it which, on the basis of the hypo-hyper-elastic theory, can be explained by the fact that under the conditions stated for Fig. 12, the spring *S* would execute a series of oscillations during the momentary period of instability.

In all of the above discussion the stress-strain diagrams have been based on the original cross-sectional area of the steel piece they represent. Actually, a steel specimen "necks" or decreases locally in diameter, and in doing so becomes considerably harder

[4] W. E. Dalby, Proceedings of the Royal Society of London, vol. 86A, 1911–12.

at that particular section. The diagrams discussed do not take into account this additional elastic energy which is of the hyper type. Fig. 13 gives a stress-strain diagram based on the actual diameter as it varies to the point of fracture, and on it the correct hyper-elastic area *abcda* is shown. Comparing this more exact area with the hyper-elastic area *efgde* by the diagram in current use, we see that it is very much greater.

HYPER-ELASTIC HYSTERESIS

Hyper-elastic hysteresis is a lag in deflection (or strain) following the application or release of a load (or stress) within the hyper-elastic range. Fig. 14 shows the manner in which the hyper-elastic hysteresis loop is formed. Area $OP_E e$ represents hypo-elastic energy, area $gfP_{E_1}eb$ the hyper-elastic energy, and area $aP_{E_1}b$ the sum of the two or the total elastic energy thus far expended. In releasing all load one would expect on first thought that since all elastic action is retroactive, the unloading would proceed along line $P_{E_1}fda$, but such is not the case. Instead the unloading follows the line $P_{E_1}ec$, which indicates that with the exception of elastic energy *adc*, the hyper-elastic energy is lost as hysteresis—due no doubt to self-interference of grains in the aggregate, and to the simultaneous action of hyper-elastic action with plastic flow during loading. Only the hypo-elastic energy which was developed freely on loading is recovered. Reloading therefore proceeds along the line cdP_{E_1}, the new proportional limit being higher than the original P_E. Area $cdPe_1ec$ is called the hyper-elastic hysteresis loop, and it will be noted that with the addition of area *adc* it is equal to the hyper-elastic energy $gfP_{E_1}b$. The energy represented by area *adc* is not lost, its particular function being to raise the proportional limit from P_E to P_{E_1}.

To summarize, we can say that most of the hyper-elastic energy is lost upon cyclic unloading and loading, the loss being equal to the hyper-elastic hysteresis loop. The remaining small amount of hyper-elastic energy remains as "trapped" or stored stress which manifests itself by a higher proportional limit. A material in this state is said to be overstrained. In certain metals such as zinc, wherein the percentage of plastic energy to elastic energy is high, the deformation *Oc* in Fig. 14 (*a*) creeps back to *Oa*, thus releasing the "trapped" energy *adc* and lowering P_E to *f*. This creeping back, which is called "elastic after-effect," does not take place in all metals immediately upon unloading, but requires, in the case of soft iron, heating to a temperature of 212 deg. fahr., and in the case of certain steels and phosphor bronzes, to a temperature of about 500 to 600 deg. fahr. Likewise, upon loading, point P_E may extend or creep to *h*, and, as in the case of elastic after-effect, may for certain metals require a given temperature to accelerate this motion. The latter movement is usually designated as "creep." This time effect in connection with the hyper-elastic hysteresis loop is to be expected, since it is derived from hyper-elastic energy, which, as we have seen, is developed simultaneously with plastic energy, whose characteristic depends upon the factor of time. As a matter of fact, under proper conditions of cyclic speed certain materials might not show any hyper-elastic hysteresis loop, even though the true proportional limit be exceeded. This would be due to two reasons: namely, sufficient internal heat would be developed to accelerate creep and elastic after-effect, and there would be a normally low plastic-flow limit. At this point it should be repeated that viscosity, and hence plastic flow, increases with temperature as well as with pressure, which accounts for the development of greater plastic energy (greater ductility) at elevated temperatures. On the other hand, elastic energy, particularly of the metals, decreases with temperature due to a reduction in atomic attraction brought about by the increased thermic vibration.

In Fig. 14 (*b*) several more cycles of loadings and unloadings have been added, and in each case the new and higher proportional limit has been exceeded. In each loop the corners O_1, O_2, O_3, e_1, e_2, and e_3 have been rounded, which is more in accordance with experimental evidence and which is due to the statistical character of properties of crystalline metal aggregates. From these progressive cycles or hysteresis loops it is evident that—

1 The hypo-elastic energy remains constant.
2 The hyper-elastic energy increases.
3 The hyper-elastic hysteresis loop increases in area.
4 The "trapped" stress energy increases, hence the possible elastic after-effect and corresponding creep also increase.
5 The proportional limit apparently increases, but actually the proportional limit of the elastic phase remains constant.
6 Proportionality below the apparent proportional limit is, paradoxically, destroyed, although actually proportionality of the elastic phase has not fallen down.

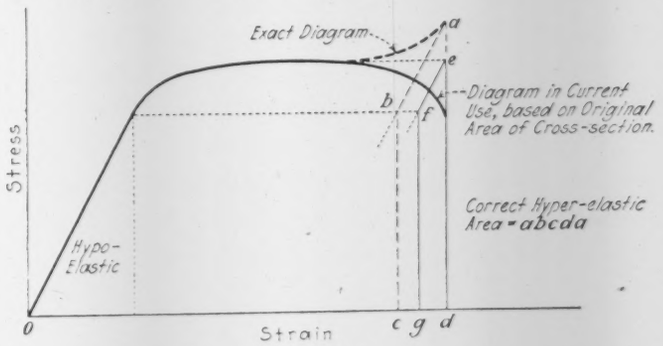

FIG. 13 STRESS-STRAIN DIAGRAM BASED ON ACTUAL CROSS-SECTIONAL AREA

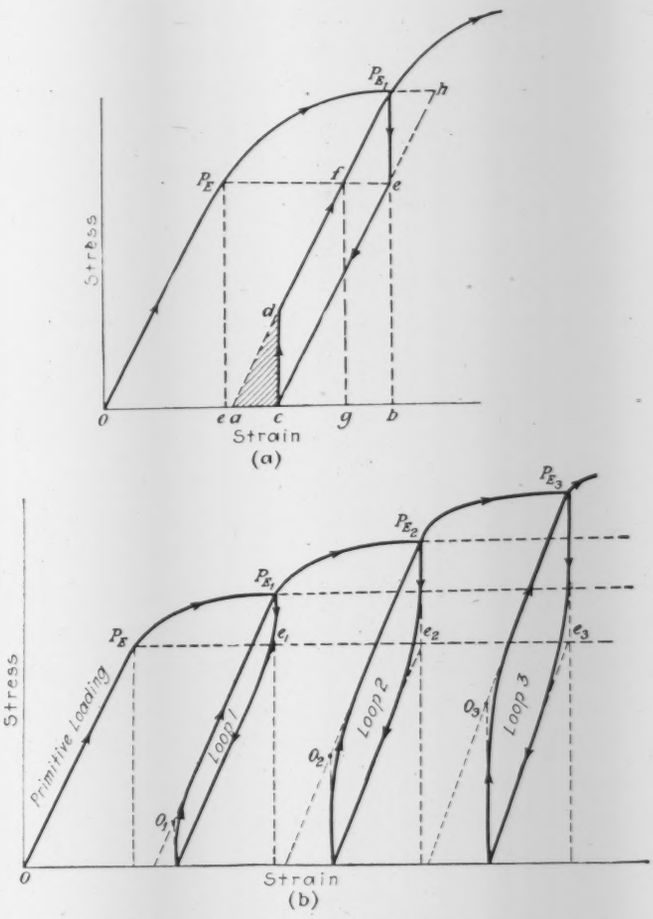

FIG. 14 MANNER IN WHICH HYPER-ELASTIC HYSTERESIS LOOP IS FORMED

Without the hypo-hyper-elastic conception, one is apt to draw vague and erroneous conclusions as to how an overstrained metal acquires its additional strength or hardness. But from Fig. 14 (*b*), it is easily seen that the raising of the proportional limit is due to the "trapping" of some of the hyper-elastic stresses. We shall see later in the consideration of full or reversed cycles how the additional hyper-elastic energy is acquired, and this it is essential to know, since we have seen that the hysteresis loops are derived from this source. Figs. 15 and 16 show two diagrams obtained experimentally;[5] in the former no elastic after-effect is in evidence,

[5] W. E. Dalby, Proceedings of the Royal Society of London, vol. 86A, 1911–12.

while in the latter it is quite pronounced, which is to be expected in view of the dominance of the plastic phase in zinc.

Fig. 17 consists of a series of complete or reversed-stress cycles which show, as it was shown experimentally many years ago by Bauschinger, that as the proportional limit is increased in the positive direction it is correspondingly decreased in the negative direction. Thus $+P_{E1}$, $+P_{E2}$, and $+P_{E3}$ represent positive increases, and correspondingly $-P_{E1}$, $-P_{E2}$ and $-P_{E3}$ represent negative decreases. In the primitive cycle $(-P_{E1})O(+P_{E1})$, so called

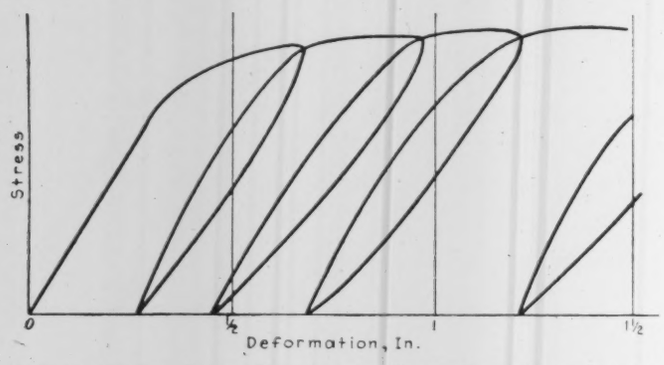

FIG. 15 ACTUAL STRESS DIAGRAM FOR BRASS OBTAINED BY DALBY

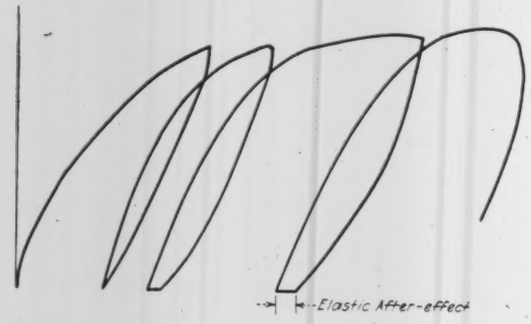

FIG. 16 ELASTIC AFTER-EFFECT IN ZINC ACCORDING TO DALBY

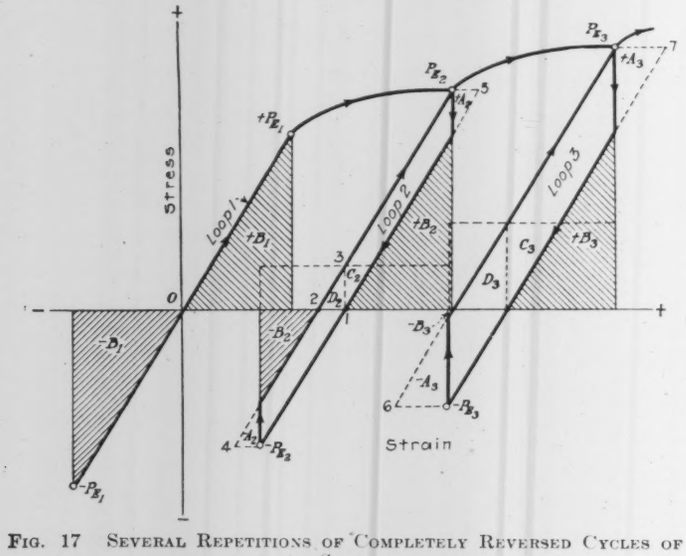

FIG. 17 SEVERAL REPETITIONS OF COMPLETELY REVERSED CYCLES OF STRESS

because it presents a piece of metal in a perfectly unstrained or virgin state, the hysteresis of the hyper-type is zero within the limiting stresses $+P_{E1}$ and $-P_{E1}$. The first positive overstraining of the metal during its history raises the positive proportional limit to $+P_{E2}$ and decreases the negative proportional limit to $-P_{E2}$. It also creates the hyper-elastic hysteresis loop in the following manner: In the primitive state the hypo-elastic area $+B$ equals $-B$, while in the first overstrained-state area $+B_2$ equals $+B_1$ of the primitive state, but hypo-elastic area $-B_2$ is less than $-B_1$. A study of the diagram shows that the balance of hypo-elastic energy $-B_1$ minus $-B_2$ equals the positive part of the acquired hyper-elastic loop less the areas D_2 and C_2, which, as we have learned previously, is equal to the hyper-elastic energy. Areas D_2 and C_2 represent the energy due to trapped hyper-elastic stress and correspond to $-A_2$ and $+A_2$, respectively. Likewise with the third overstrained state, more of the negative hypo-elastic energy is converted into hyper-elastic energy, which in turn is lost as hysteresis and trapped stress in virtue of which the positive proportional limit is further elevated and the negative proportional limit is further depressed.

From the above principles the very important deduction can be made that the total hypo-elastic energy of a material in the primitive or virgin state is the total elastic energy which that material is capable of developing in a complete cycle of stress. Hyper-elastic energy does not represent a gain since it is practically all lost in the hyper-elastic loop. Hyper-elastic stresses which give rise to hysteresis are trapped due to the fact that slip or plastic action is irreversible until the slip stress, i.e., the negative proportional limit, is reached in the reverse direction. This dependence of irreversibility of slip on negative hypo-elastic action is the under-

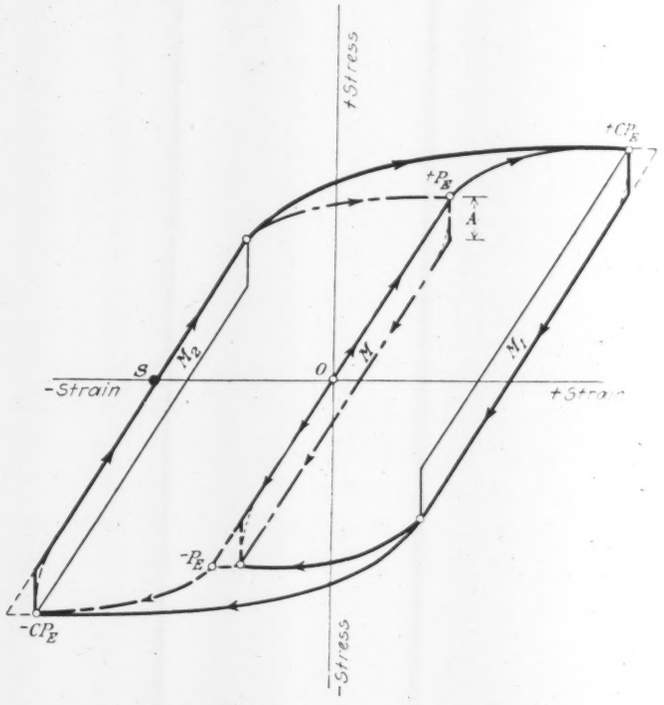

FIG. 18 OSCILLATION OF THE HYPER-ELASTIC HYSTERESIS LOOP

lying reason of all hyper-elastic hysteresis. In confirmation of this, it has been found by many experimenters[6] that a definite frequency of stress repetition will obliterate hyper-elastic hysteresis, which can be accounted for by the fact that sufficient heat is developed on the slip surfaces to reduce the viscosity materially and enable instantaneous creep and elastic after-effect to take place. For instance, Gough[7] finds the fatigue endurance limit by determining the "kinetic" proportional limit, which can only be accomplished by obliterating the hyper-elastic hysteresis loop caused by overstraining a metal to the "commercial" proportional limit. The "kinetic" proportional limit determined while the metal is under quick repetitions of stress is the true proportional limit, and since the true proportional limit is the stress starting the first slip, it is also the fatigue endurance limit. Due to lack of sensitivity in the static determination of the proportional limit, the metal is overstrained, and the "commercial" figure thus obtained represents a limit higher than the primitive figure.

Fig. 18 shows a closed cycle, but not strictly of the hyper-elastic type since it also includes the area of plastic energy. Nevertheless it is a hysteresis loop that would be obtained in stressing a metal between its commercial proportional limits $+CP_E$ and $-CP_E$. The

[6] G. H. Keulegan, Statical Hysteresis in the Flexure of Bars, Technologic Paper No. 332, U. S. Bureau of Standards.

[7] Gough, H. J. Improvements in Methods of Fatigue Testing, *The Engineer*, August 12, 1921.

true proportional limits $+P_E$ and $-P_E$, or stresses of first slip, are the endurance limits, from which it follows that the endurance limits are always to be found within the commercial proportional limits to different degrees, depending upon the nature of the metal and upon the sensitivity of testing equipment. There is considerable information to be gleaned from this diagram in conjunction with experimental data as regards fatigue, but the author will have to make this the subject of a future paper. Attention will be directed at this time, however, to an important fact shown

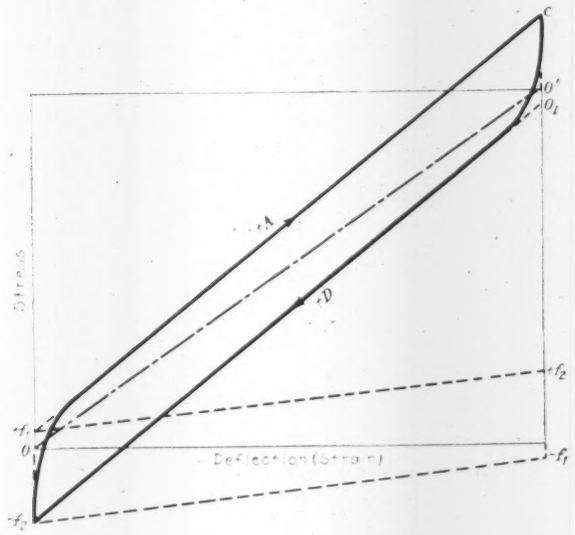

FIG. 19 HYPO-ELASTIC LOOP

by the diagram, namely, that the hyper-elastic hysteresis loop M oscillates between the extreme positions of M_1 and M_2, the amplitude either side of the zero position being equal to A.

HYPO-ELASTIC HYSTERESIS

We have seen in the foregoing discussion on hyper-elastic hysteresis that under a relatively high frequency of cyclic repetition a state is set up which obliterates this type of loop. It will be shown now that there is another entirely different type of hysteresis which is always encountered, although to an extremely smaller degree, within the hypo-elastic range of stress, i.e., the true proportional limits. High frequency of cyclic repetition will not obliterate hypo-elastic hysteresis, and time in no way effects the size of the loop obtained. Loop size seems to vary only with stress range and nature of material. For this and other reasons, experimental[8] and mathematical,[9] the cause of hypo-elastic hysteresis seems to be one of internal friction of the solid instead of the viscous type. It therefore takes the shape of the loop shown in Fig. 9, but instead of leaning toward the lenticular shape as this does, it tends somewhat toward the viscous as shown in Fig. 4, but for a reason entirely different from that of time effect. Kimball[8] has shown that this type of internal frictional energy varies as the square of the amplitude of stress, which would support the previous contention and would also give the logarithmic rate of damping of free oscillations shown in Fig. 8, which coincidence with viscous damping has led many analysts excepting Lees[9] astray. It is the author's feeling that the internal friction causing hypo-elastic hysteresis takes place at the grain boundaries due to their tendency to rotate. The couple N shown in Fig. 10 (*a*) which must be applied in order to maintain equilibrium is the normal to this force of friction, and since this normal must increase with stress range, the friction must increase also, most probably as Kimball has found. Fig. 19 shows a hypo-elastic loop constructed on this basis.

Fig. 20 shows a combined hypo-elastic and hyper-elastic loop. The cyclic state (rapid repetitions of stress) reduces the hyper or outside loop $OPO'P'O$ to the inner or hypo loop $OMO'M'O$.

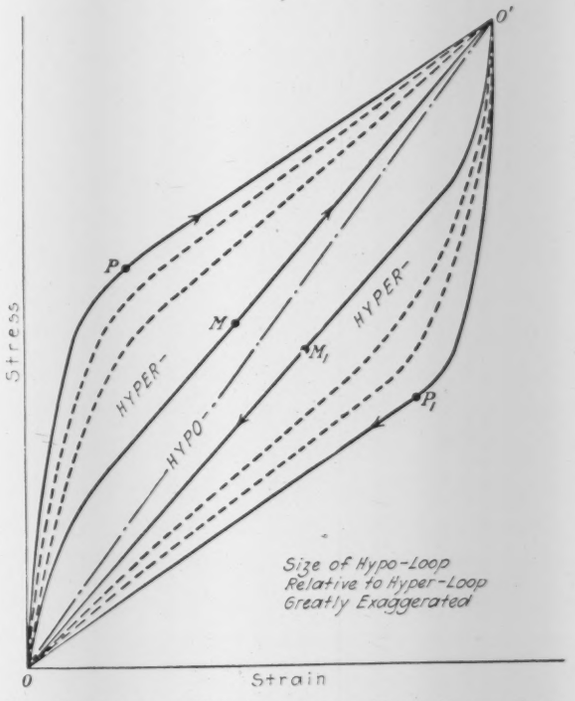

FIG. 20 COMBINED HYPO-ELASTIC AND HYPER-ELASTIC LOOP

In determining the fatigue endurance limit by the temperature method it is found that there is a very slight and uniform increase of temperature up to the endurance limit, and that at this stress there is a rapid increase. Might not this uniform slight increase be due to the friction of hypo-elastic hysteresis, which is relatively small, and the large increase be due to heat generated by the crystalline slip of hyper-elastic hysteresis, which is relatively large?

[8] A. L. Kimball and D. E. Lovell, Internal Friction in Solids, Trans. A.S.M.E., vol. 48 (1926), p. 479.

[9] S. Lees, *Philosophical Magazine*, 6 series—44, July–Dec., 1922, p. 374.

A Description of the Henry Ford Trade School

Particulars of a Successful, Self-Supporting Trade School with 1700 Students Who Work Two Weeks Out of Three in the School Shop, Devote the Third Week to a Course of Study, and Who Each Receive from $450 to $1000 Annually

By FREDERICK E. SEARLE[1]

EDUCATION is the fitting of a person for life. It is not merely the assimilation of vast numbers of words. Neither is it the parrot-like acquiring of theories. Fundamentally, it is training in how to use the faculties with which a person is born and the knowledge which he may obtain.

Paper does not and cannot take the place of practice. Theory cannot take the place of fact. Hypothesis cannot take the place of action. An educated man is one who can *do* things as well as *know* things. He cannot count himself educated unless he can make effective use of what he knows.

The digestion of dictionaries has too long been mistaken for learning. A book may contain all the knowledge in the world and still be useless if no one opens it. A man may possess all the learning of the human race and still be useless if he does not make practical application of that learning. True education enables a man to accomplish things. When it fails in that it fails in everything. Nowhere on earth is there any one so pitiful—or so futile—as the "educated" man who cannot make his own way in the world.

And yet schools have been educating people away from life. They have stressed the knowing and neglected the doing.

The greatest single flaw in modern formal education is the neglect of the hand in favor of the head. Pupils are told how others do things, but they are not asked to do these things themselves. They hear of work in the abstract. They may even see it but they do not *do* it. Their ears and their eyes and their tongues are employed. But their hands are left idle.

Yet the hand is considered by many the most important of all implements of the mind. It is the chief pathway to the brain. It conveys more impressions to, and more expressions from, the brain than all other agencies combined. It has been estimated by competent educators that 90 per cent of all elements of education are communicated to the brain by the hand. It is the symbol of usefulness, the instrument of usefulness. The turning of virtually all acquired information to practical purpose must come through the hand. Contact and action are attributes of the hand. These, and accuracy, initiative, judgment, ability, effort, ingenuity, discipline.

Disadvantages of Modern Education

Modern education has a tendency to withdraw youth from life until twenty years or more have passed. These twenty years are devoted to talk—to discussions of what others have done. Work seems to the pupil far distant and unreal. It is something to be talked about, but not accomplished. Then, at the end of the twenty years, he is plunged abruptly into the world. There for the first time he learns that the person who can do things is rated high above the one who can only talk about them.

The remedy for all this must lie in coördination of hand and head in the schooling of youth. Energy must be added to education. Work cannot be found in books. It is found only in experience. And work comes through the hand. Books can tell a pupil how others have worked, but his own hands tell him how he himself has worked.

This paper is not a plea for manual training. It is not even a defense of much that passes for technical training. It is merely a description (brief, and therefore inadequate) of one school where the coördination of hand and head has been attempted with justifiable results—a school where pupils read *how* a thing is done and *why* it is done, *and then do it.*

[1] Superintendent, Ford Schools.

Contributed by the Committee on Education and Training for the Industries for presentation at the Spring Meeting, White Sulphur Springs, W. Va., May 23 to 26, 1927, of The American Society of Mechanical Engineers, 29 West 39th Street, New York. All papers are subject to revision.

The Henry Ford Trade School is fortunate. It has an industry for a textbook and a steel mill for a laboratory. It has instructors who have done what they teach. It has definite aim (which, sadly enough, many schools lack). It is equipped to produce useful articles, not playthings to be discarded. It has class rooms for theory and shop rooms for practice—and theory and practice go hand in hand in the instruction of the pupils.

What a boy learns in the class room has direct bearing on what he learns in the shop. What he learns in the shop fills out what he learns in class. The one is explanation, the other demonstration. And the demonstrating is done by the boy himself. He learns a thing by doing it, not by hearing others tell of it.

The school is intended for youths less fortunate than their fellows—not to "help" them in the sense of the professional philanthropists, but to help them help themselves. Its purpose is not to establish records, nor to set itself up as a model for the world to follow. It is not a school of "picked" youths, if by picked one means selected for especial brilliance. The boys are selected for their own needs, not the needs of the school. Orphans make up 10 per cent of the school's enrolment. Forty-five per cent are widows' sons. Fully 80 per cent contribute to the support of themselves or of others at home. The boys are neither remarkably brilliant nor markedly dull. Taken as a whole they average pretty closely to the type of boys found in other schools of similar age range. Yet they progress at a rate much faster than that of other schools. Their aptitude not only along mechanical lines but in the higher technical and even cultural branches substantiates the belief that the basic idea of the school is fundamentally sound.

By no stretch of the imagination is the school a charitable institution. Charity makes drones. The boys at the Henry Ford Trade School are not drones. They are workers. And they are happy, working. The school teaches them self-reliance. It teaches them sturdy independence. They are under no obligations when they come, and they are under no obligations when they leave. They work as they learn, and they earn as they learn.

Cardinal Principles Followed in the Henry Ford School

The Henry Ford Trade School was established in October, 1916, with six pupils and one instructor. There are now 1700 pupils and 125 instructors, with a waiting list of more than 5000 local boys eager for admittance. At the time the school was founded, three cardinal principles were laid down. They were:

> Each boy to be kept a boy and not made a premature workingman.
>
> Industrial training to keep pace with academic instruction.
>
> Pupils to be trained by making products of actual use, not articles to be thrown away.

These principles have been closely followed in the development and expansion of the school. Underlying each activity has been the desire to train the boys both to think and to act for themselves.

Youths are admitted to the school between the ages of twelve and fifteen. When they become eighteen they are automatically graduated from the Junior Course, but they may continue their studies in the Senior Course until they are twenty. At this time, or before, they are offered positions with the Ford Motor Company. The academic training in the Trade School corresponds to that of an accredited high school. The shop training is believed to exceed that of any similar school in the world.

Upon entering the school each boy receives a cash scholarship amounting to $7.20 a week. This is paid him on the fifth and twentieth of each month, and continues through a vacation of three weeks in the summer and one week at Christmas. A report

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covering the pupil's work is prepared each six weeks. If this report shows that the boy is applying himself, his scholarship is increased 40 cents a week. When the $10 rate is reached, the character of work produced in the shop is also taken into consideration. By the time an attentive youth is seventeen, he should be receiving a scholarship of $18 a week, which is the maximum for a boy until he is eighteen.

Thrift is considered one of the most important of all lessons taught at the Henry Ford Trade School.

To encourage boys in developing it, the school awards each pupil an additional $2 a month as thrift money. This must be deposited in some bank and kept there as long as the boy remains in school. Bank books must be submitted regularly to show that the money has not been withdrawn, but it is the exclusive property of the boy and is turned over to him without restriction when he leaves the school.

Each noon a hot luncheon is furnished the boys. The three items—scholarship, thrift fund, luncheon—make the minimum amount received by the pupil $450 a year, and the maximum, $1020. Under the present enrolment this amounts to more than $1,000,000 annually.

The main building of the Henry Ford Trade school adjoins the Highland Park Plant of the Ford Motor Company. A branch of the school has recently been opened at the Fordson Plant, with accommodations at present for 150 pupils.

While complementing each other, the two branches of instruction are kept entirely separate. Each boy devotes one week of study to the class room, and then two weeks to the shop.

THE ACADEMIC CURRICULUM

During the academic week the boy attends class work only. Throughout his course he attends classes in

English	Auto Mechanics
Mechanical Drawing	Mechanical Science
Civics	

The following subjects are one-year courses and are taken in the order given:

Shop Arithmetic	Qualitative Analysis
Algebra	Physics
Commercial Geography	Quantitative Analysis
Elementary Chemistry	Metallurgy
Geometry	Metallography.

The work in drawing, chemistry, and mechanical drawing is of a more advanced grade than that given in high schools, and boys have been given advanced credit in college for these subjects.

Boys who wish to attend college take history and foreign languages in some other school. Many of the boys are doing this.

All boys come to school at 7:30 each morning for five days a week. Those in the shop are dismissed at 3:55 p.m. Those in class leave school an hour earlier and are credited with an eight-hour day.

The school shop is separate from the Ford Motor Company and covers three acres of floor space. On one floor is a series of rooms totaling a length of 1400 ft. with an average width of 70 ft. In this shop are hundreds of the finest machines of many types. The total equipment is valued at more than $1,000,000. In this shop there are 18 departments, and two men spend their entire time moving boys from one department to another as fast as they have completed the requirements. An accurate record is kept, not merely of the department in which a boy has worked, but also of the various operations he has performed in each department.

SHOP COURSES

A new boy is always placed on a machine set to do one simple operation, and he is kept on this work until he becomes in a measure accustomed to machinery. He is then transferred to work on a shaper, lathe, milling machine, grinder, and bench. As far as possible boys are also given an opportunity to work in the following departments:

Forge	Foundry	Car Repair
Die	Sheet Metal	Wood and Metal Pattern
Gage	Nickel Plating	Carpentry
Hardening	Valve Repair	Tool Repair.

In the shop each job is done on a work order. An expert estimates how many hours it would take a skilled mechanic to complete the work. An accurate record is kept, and boys and instructors try to come within the estimate. Over a period of years the percentage of excess time required has varied greatly, rising to 60 per cent and sinking to 13 per cent for the month of September, 1926. The average is about 25 per cent more than the time required by skilled men.

The spoilage averages 4 per cent, which is about twice that of the average tool room. No work is done merely for practice. Everything the boys make is to be used unless spoiled. Even in the class, wherever it is possible, practical work is done by the boys. Instruction in mechanical drawing is given during the entire course, and many of the older boys, under the guidance of instructors, do development work for the Ford Motor Company for which the school receives credit.

At eighteen the boy enters the Senior Course. He works in the shop eight hours each day and attends class work in advanced drawing and mathematics four hours each week. His rate is gradually increased to $30 a week by the time he is nineteen, and may reach $40 a week when he is twenty. At twenty or before he is offered a position in some department of the Ford Motor Company.

The activities of the boys are by no means limited to the class room or the shop. An orchestra shares musical honors with a glee club. The *Artisan*, a creditable school paper edited by the boys, is published twice a month. A radio club delves the mysteries of the ether. A dramatic society has put on several worth-while shows. The Exploration Club is a social organization through which the pupils come into direct, friendly contact with their instructors. Inter-class athletic games are arranged, but no teams represent the school.

THE SCHOOL SELF-SUPPORTING

The annual value of material produced in the shop by each boy is approximately $1000. Thus the income from work done for the Ford Motor Company is about $1,700,000. This is sufficient to pay the students' scholarships, salaries to instructors, and all the upkeep of the equipment. It does not, however, pay interest on the investment in buildings and equipment.

Educational work in the Ford Motor Company is not confined to the Henry Ford Trade School. There is the Ford Apprentice School, where 1500 men between the ages of twenty and twenty-five are trained as skilled mechanics. The course covers a period of three years. The pupils are not segregated. They work along with other men in the various departments, but a foreman in each tool room moves them from one machine to another as fast as they master each job. For the first two years of the training period the apprentice must attend one class each week in mechanical drawing and mathematics. The class periods come either before or after their regular working hours, and for them they receive no pay. The starting rate for apprentices is $6 a day. This is gradually increased to $8 a day by the time the course is completed.

In addition to the regular classes, courses in electricity, metallurgy, and metallography are conducted for the benefit of those employees whose work makes information in these subjects helpful. The mathematics and drawing courses were written in the department, and include not only a general training in these subjects but a large number of actual problems peculiar to tool and die making.

SPECIAL STUDENTS FROM ABROAD

The Ford Service School has an enrolment of 300 special students from more than thirty different countries. Many of them are graduates of colleges and technical schools who are desirous of serving their own country by studying those things in American industry which can be used to advantage in their homelands. It is partly as a courtesy to the nations they represent, and partly in the hope that they may later prove an asset to the company, that the Ford Motor Company has placed them in training.

The school is recognized as an educational institution by the United States Department of Labor, and the foreign students are permitted to enter the country on student visas without regard to the quota allowed their nation. The course requires two years. The men are placed in a few manufacturing departments of the

company, but most of their time is passed in the various assembly and repair divisions.

At the present time delegations are in the school from Armenia, Australia, Brazil, Canada, Chile, China, Colombia, Costa Rica, Cuba, Czechoslovakia, Egypt, Denmark, Finland, France, Germany, Guatemala, Holland, India, Japan, Jugoslavia, Mexico, Norway, Panama, Persia, Philippine Islands, Poland, Porto Rico, Portugal, Russia, Scotland, Spain, Sweden, Switzerland, Turkey, and Greece. The rate for each student is $6 a day as long as he remains on the course.

Coöperative Students from Different Colleges

More than 200 coöperative students are also received by the Ford Motor Company from different colleges, the larger groups coming from the University of Detroit, the Detroit School of Technology, the University of Cincinnati, and Antioch College. The students are placed in the tool rooms and special departments. Those who are assigned to a single department may receive more than the minimum wage; those who are shifted through the various departments remain on the $6 rate. Many of these students remain with the organization after completing their college courses, the special technical training proving of great value to them.

It is impossible at this time, of course, to estimate accurately the significance of the Henry Ford Trade School. That it is of value both to the pupils and to the company is apparent. The oldest of the graduates are now only twenty-six, but already they are forging ahead of their fellows who have not had the benefit of the special training. And from the graduates of the school the company expects to draw many of its higher executives of the future.

But the true worth of the school will be measured not so much in terms of those who have direct contact with it, as in the effect it will have on educational measures at large. Here, for almost the first time, the coördination of head and hand in the instruction of youth has been attempted on a large scale. Students are not alone taught how to acquire knowledge. *They are taught how to use it.*

Industry in the past quarter-century has progressed farther than during any other thousand-year period in all history. Education has failed to keep pace. The Henry Ford Trade School is an attempt to lessen the breach between the two.

The Relation of Building Design to the Manufacturing Process

Broad Principles Governing Building Projects—Special Features of Building Design, Assuming Process to Have Been Perfected—Descriptions of Buildings for the Manufacture of Soap, Linoleum, Plate Glass, and Textiles, and for Publishing Magazines and Newspapers

By CHARLES P. WOOD,[1] NEW YORK, N. Y.

THE purpose of a manufacturing building is to protect equipment and stock and provide working quarters for employees. The ideal building would have all its features developed just enough to insure efficient operation. To get the building right requires knowledge of the relative values of the features that make up a complete building. Certain features are more important for one kind of operation than for another. Therefore the manufacturing process, which is the source of income, should be the governing factor in the building project.

The first cost of a building may be so low that repairs and depreciation will make the actual cost higher than that of a building which requires a larger initial expenditure. At the same time there is danger of investing more in buildings than the company can afford.

Money saved in a building operation is made available for more profitable investment elsewhere. To spend more than seems at first necessary is sometimes justified by advertising value, provision for future expansion, adaptability for other than the original occupant, etc. Before the design is started the policy of the company should be made clear, as it affects financing, future growth, and possibility of changes.

In brief, an economic study should precede the building plans.

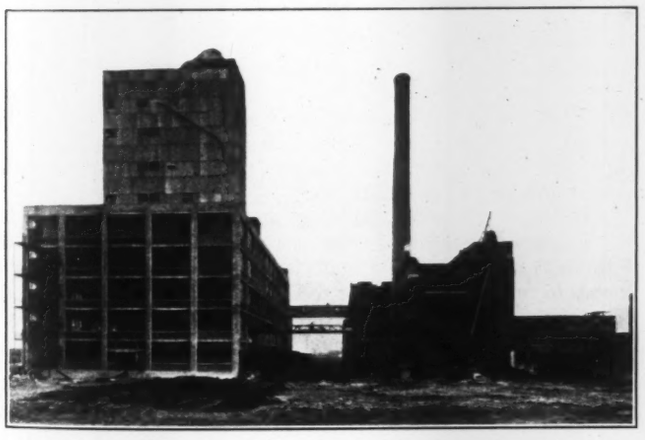

FIG. 1 WASHING-POWDER AND SOAP FACTORY

The value of the land, the earning capacity of the business, the advertising value of the location and of the building's appearance—these and other features outside the process have much to do with the character of the building. A building which suits the process but which can be converted to another purpose, for instance, is a better investment than one which is not adaptable for a change of occupancy. This is only common sense, but it is something worth repeating as we think of the number of expensive buildings offered for sale around the country at a fraction of their first cost because they were really suitable for only one thing and that thing did not succeed at that place. It has been found that new discoveries, inventions, and changing conditions require changes in a process from time to time. Some of these changes make necessary the complete rearrangement of a department or of the entire factory. If the building has been too highly specialized in the beginning, the cost of alterations and additions may be out of all proportion to the benefits already derived from the original design. No matter how clever some special feature may be, it should not be allowed to block future developments unless it promises to pay for itself in the meantime. While it may pay to invest $15,000 to save one man's time and $10,000 to save one girl's time, the operation will have to continue for ten years before the investment will be protected. Idle space results from "playing safe" whenever in doubt—a common fault which starts as a frozen asset and develops into a liability.

So far we are dealing only with the broad principles that govern the building project before the special requirements of the process have been considered. Having determined the limits within which the work must be kept, the process becomes the first thing to lay out, and then the building remains to be put around the process.

FIG. 2 TYPICAL FLOOR OF MAIN BUILDING SHOWN IN FIG. 1

Provision for the comfort of employees may be a matter of sentiment in some cases, but generally it pays. The demand, in this respect, varies with the locality and the kind of people who work in the factory. A comfortable place to work, with good light, increases production and improves quality in a silk mill or machine works, for instance, while the same thing would be extravagant in a fertilizer factory. Competition for the better class of labor gives the advantage to the factory that has clean lavatories, lockers, dressing rooms, and rest rooms. Skilled workmen, male and female, prefer to change their clothes at the factory. Many intelligent girls seek employment when they are sure they can keep their street clothes in a safe place and look well on their way to and from work. The lunch room or cafeteria is a problem that depends upon the number and class of workers and the location of the plant. Habits and preferences have to be studied before the solution can be found. In general, experience has shown how far to go with expenditures for comforts before having to make a charge against such facilities as non-productive.

Good appearance has to be sacrificed to utility, in the main, but a well-proportioned building can be made to look well without spending money for ornamentation. Of the two, proportion gives a more pleasing effect than ornamentation. Good engineering demands a sense of proportion beyond the strict requirements of the process, and here is where the client gets something extra for his money when he retains an engineer that knows his business.

This paper will be devoted to special features of building design on the assumption that the process, in each case, has been per-

[1] Lockwood, Greene & Co., Inc. Mem. A.I.E.E.

Contributed by the Materials Handling Division for presentation at the Spring Meeting, White Sulphur Springs, W. Va., May 23 to 26, 1927, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, 29 West 39th Street, New York. All papers are subject to revision.

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fected. It will be understood, then, that the sequence of operations has been worked out to eliminate unnecessary handling, to take advantage of movement by gravity, and to use the simplest and most efficient conveying systems where such appliances are needed. Furthermore, the prevailing standards for safety, insurance, and building codes are taken for granted. Buildings already constructed for a variety of purposes will be described, so the examples taken may be accepted as actually in service rather than as theoretical schemes of doubtful value. The photographs presented were taken before this paper was contemplated.

The buildings to be described cover plants for making soap, linoleum, plate glass, rayon, silk and cotton goods, and for printing magazines and newspapers.

WASHING-POWDER AND SOAP FACTORY

Fig. 1 shows the exterior of the complete plant. The main building with its tower is at the left. This is a general manufacturing building. The tower is designed for a special powder process. The first and second floors are used for the receiving, shipping, and storage of raw and finished products. Soap is pumped from the kettle house (center background) to the third floor where it is cooled, cut into slabs, and taken to the fifth floor where it is cut into cakes, pressed, dried, and wrapped, and then sent by spiral chutes to the first and second floors. The fourth floor is devoted to another kind of soap.

The glycerine building is shown in the background, to the right of the kettle house, and the power house in the center foreground. Communication between buildings at upper levels is provided by bridges.

This plant is designed for future expansion as follows: Washing powder, toward the front at left; soap, toward the rear at left; power house, toward the front at center; kettle house, toward the rear at center; glycerine house, toward the rear at right; tank field, toward the rear.

Fig. 2 shows a typical floor of the main building before occupancy. The construction is reinforced concrete, flat-slab, designed for a floor loading of 250 lb. per sq. ft. The floors are of granolithic cement treated with a patented hardener. The bays are 20 ft. by 20 ft.

Fig. 3 shows the packing department on the third floor of the main building. A distributing belt conveyor runs over the packaging machines. After this picture was taken, belt conveyors were installed to take filled cases from the end of packaging machines to sealers, and spiral gravity chutes were constructed to deliver sealed packages to the lower floors.

LINOLEUM FACTORY

Fig. 4 shows a complete linoleum-manufacturing plant. Raw materials and other incoming shipments are received over the track at the right of the picture and stored in tanks and buildings shown in the background. These materials are mostly cork and filling, linseed oil, and burlap.

The process develops from the rear toward the front and left. Mixing and coating are done in buildings shown near the chimney. Then the process divides into three parts for drying plain and inlaid linoleum and for printing and drying printed linoleum. Inlaid drying is done in the two-story building in the left background. The tall section near the center of the picture houses festoon and rack driers for plain goods. The printing and print-drying department is in the central foreground of the main group of buildings. The finished storage and shipping department is at the left, as indicated by the railroad track along the full length of the building for outbound freight.

Expansion of every department can be provided without disturbing the equipment already installed. This is readily seen from the foregoing description and the figure, which shows vacant land adjacent to the principal departments.

This plant is literally built around a process that is highly specialized throughout. Trying to squeeze such a process into an ordinary building would increase production cost and impair the quality of the product.

PLATE-GLASS GRINDING PLANT

Fig. 5 shows a plant for grinding and polishing plate glass. The main section is 136 ft. wide by 650 ft. long. The construction is steel frame with wood roof having sawtooth skylights over the greater part of the roof area. The building is designed for future extensions to the rear and left of the picture. The permanent walls are concrete with steel sash, and the temporary walls corrugated siding with steel sash.

The building is equipped with cranes running its full length, and railroad sidings running into and across the building at each end. Factory offices, toilet rooms, etc. are on the mezzanine floor. The extensions, for which provision was made, are now being built.

RAYON PLANT

The manufacture of rayon by the cuprammonium process from cotton linters is divided into the following main divisions: 1, cleaning, washing, and preparation of linters; 2, bleaching linters; 3, dissolving; 4, mixing, aging, and filtering the solution; 5, spinning; 6, washing; 7, drying; 8, conditioning; 9, inspection; 10, winding from skeins to spools; 11, twisting; 12, reeling from spools back to skeins; 13, winding finished packages on cones or tubes; and 14, final inspection.

The first two steps in the process require buildings designed and arranged for handling bales of cotton linters weighing about 500 lb. each. There is a complicated steam and water piping system with corresponding drainage connections.

The dissolving, mixing, and filtering operations require a complete chemical process with tanks, refrigeration facilities for artificial cooling in summer, insulated walls and ceilings, and forced ventilation. Much of the piping is acid-proof lead.

Referring to Fig. 6, the operations described above are housed in the five-story concrete building at the right. This building is designed for heavy loads and wet processes, concrete being the material considered most suitable for such purposes.

The picture shows the complete rayon plant with the spinning and other operations which follow the chemical processes housed in a one-story steel-frame building having a sawtooth roof. The skylights face directly north to avoid direct sun. The roof is wood except in the washing department, where reinforced concrete is used for columns and roof to prevent rotting by exposure to moisture.

In the center of the picture the fan room is shown built above the washing department. Warm air is blown in at intervals at one side of the room and exhausted through ventilators in the roof and disk fans in the skylights.

Floors throughout the building are subjected to drip until after the drying stage has been passed. For this reason they are finished with acid-proof material (mastic) and provided with trenches to return chemicals for recovery

The spinning operation demands a building with good height and ventilation, and long spans between columns. These requirements apply also to other finishing departments. Spans are 80 ft. in the spinning department and 40 ft. elsewhere in the one-story section. This span is made possible by trusses immediately behind the skylights, clearly shown in Fig. 7. The sawtooth skylights are vertical because of their connection with the trusses just described. This construction insures the most rigid and economical framing for a building of this character. The wet skeins from the spinning department receive a number of baths before drying. This operation is somewhat similar to those in a dyehouse. There is a large amount of drip on the floors; warm water is used, and a great deal of vapor escapes into the room. Ventilation must be provided to meet these extreme conditions, and the construction must be such as will not be damaged by steam. Drips from the roof or sash must be prevented. This part of the building is protected by concrete construction. The skein-drying department also requires special ventilation.

Following the drying, the operations are similar to those for silk yarn, requiring good light, high humidity, and provision against condensation and drip in cool weather.

The plant herein described is on the outskirts of a small city in the South. A large number of girls are employed. To provide comforts and conveniences for the help, who spend the entire day in the plant, there are ample locker rooms, lunch rooms, shower baths, etc. in the basement which extends under part of the building as shown in the foreground of Fig. 6.

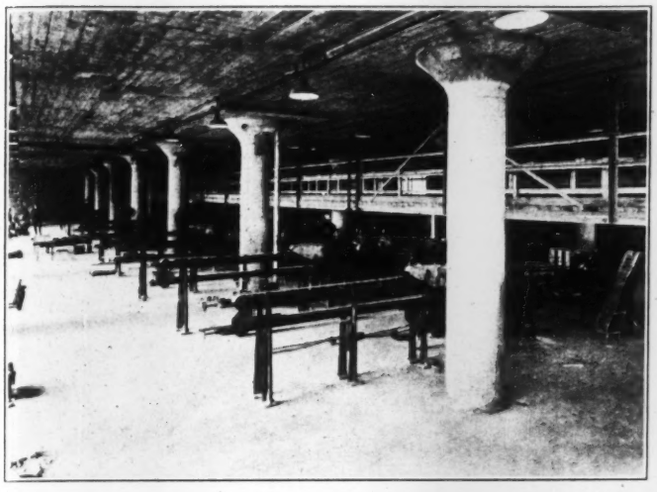

FIG. 3 PACKAGING DEPARTMENT ON THIRD FLOOR OF MAIN BUILDING, FIG. 1

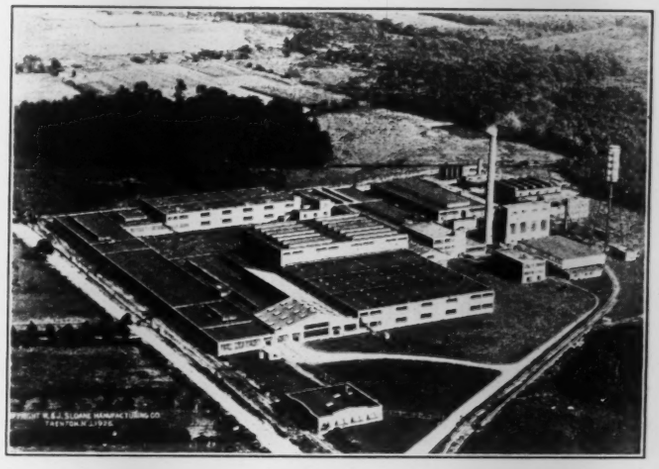

FIG. 4 LINOLEUM FACTORY

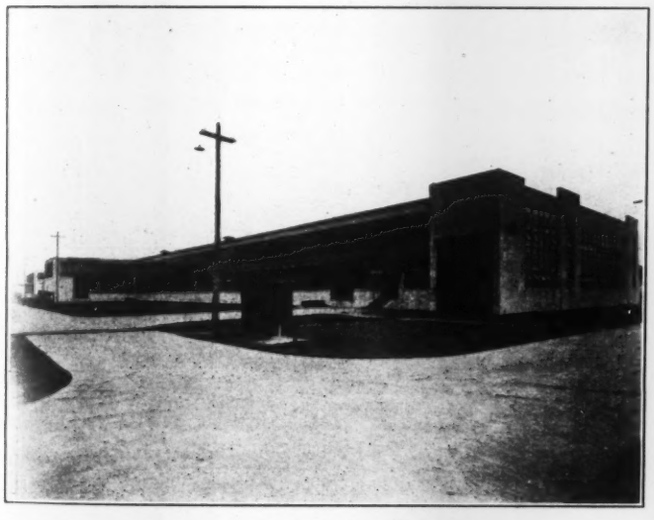

FIG. 5 PLATE-GLASS GRINDING PLANT

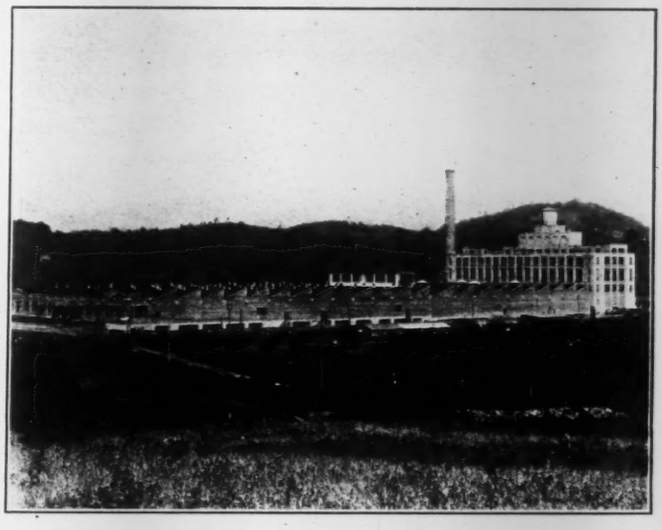

FIG. 6 RAYON PLANT

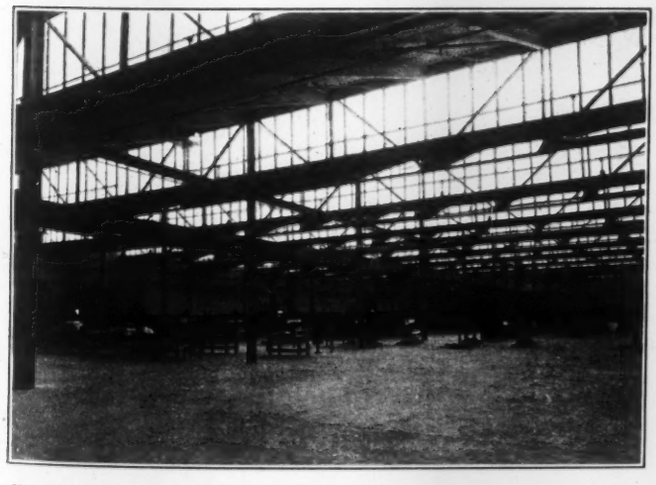

FIG. 7 TEXTILE MANUFACTURING ROOM OF RAYON PLANT, FIG. 6

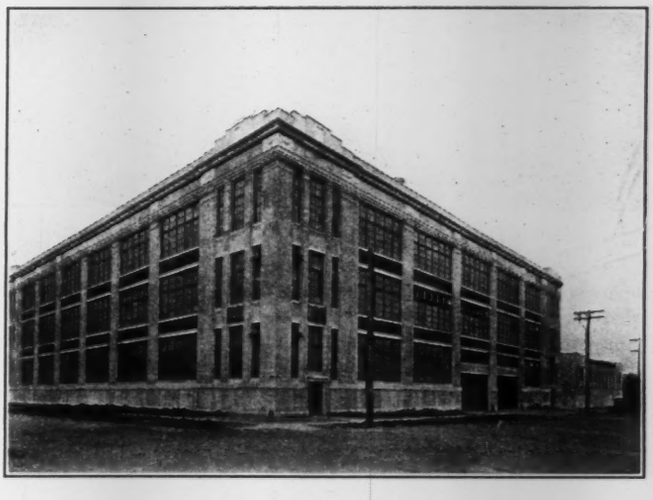

FIG. 8 SILK MILL

Two-inch cork insulation is used on the roof of the washing department and 1-in. elsewhere wherever artificial humidity is employed. Another precaution against condensation caused by artificial humidity is double glazing of sawtooth skylights with air space between glass and condensation gutters throughout.

The special provisions for insulation, ventilation, and north light insure cool rooms in summer besides the temperature regulation required by the process. The plant is equipped with central-station humidifying and cooling apparatus.

The power plant is equipped with non-condensing turbines because the exhaust is needed in the process. Boilers are set at the second-floor level. There is a trucking passage under the ash hoppers so that ashes may be discharged into iron-body trucks and hauled away. Coal is discharged into a track hopper with conveying system to crusher and to bunkers. From the bunkers coal is distributed into automatic stokers by gravity. A pipe tunnel connects the boiler house with the basement of the chemical building. The power plant is in the background behind the chemical building at the right of Fig. 6. Its location is shown by the chimney.

The pumping and filtration plant is on the river 1600 ft. from the power house. The pumps are electrically operated.

Fig. 6 shows the first unit designed for expansion into five units. The central unit is the first to be constructed. Railroad sidings at each end of the building, power plant, pumping station, sewage-disposal system, driveways, and drainage have been laid out for the ultimate requirements.

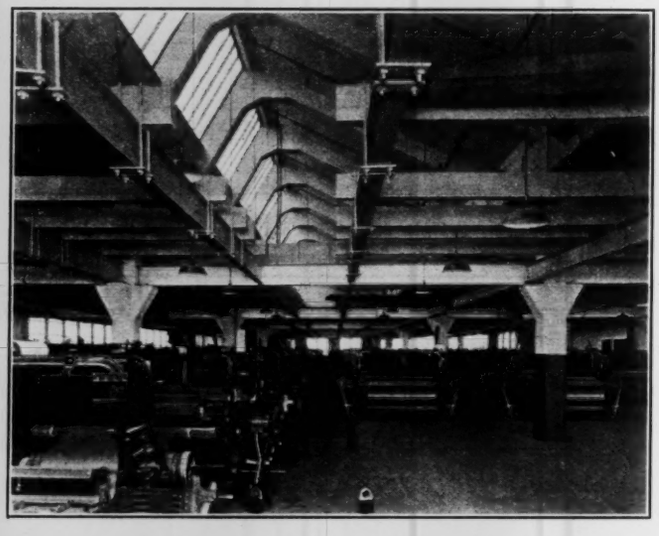

Fig. 9 Top Floor of Silk Mill Shown in Fig. 8

Fig. 11 Weave Shed of Cotton Mill

Silk Mill

The next illustration, Fig. 8, shows the exterior of a silk mill which was designed to conform with the zoning requirements in an exclusive district. The building fronts on a park, where factories having objectionable features would not be allowed. Practically nothing was spent for ornamentation of this building. The details are severely plain but the proportions are pleasing. The effect is more that of a public institution than that of a factory.

In contrast with the exterior, as shown in Fig. 8, Fig. 9 shows the interior of the top floor of the same building. The construction is reinforced concrete throughout, with sawtooth skylights. The special framing seen overhead is provided for Jacquard looms.

Fig. 10 shows the top floor of another silk mill where specially long spans between columns are required. This is accomplished by trusses behind vertical sawtooth skylights, clearly shown in the picture.

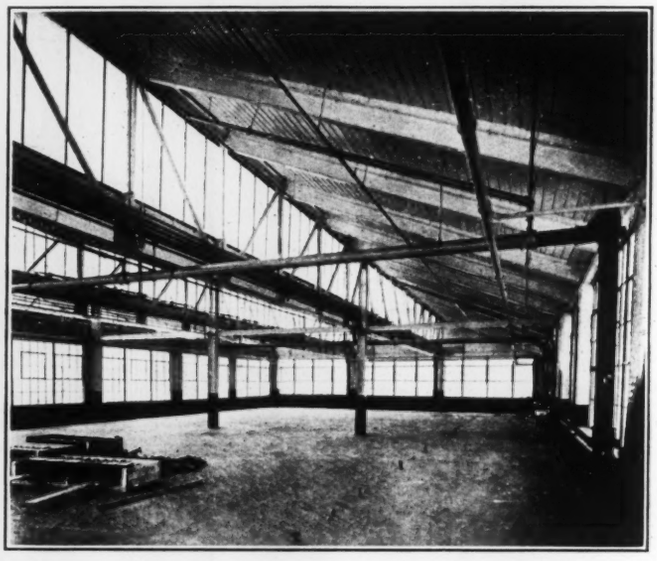

Fig. 10 Top Floor of Silk Mill, Showing Long Spans

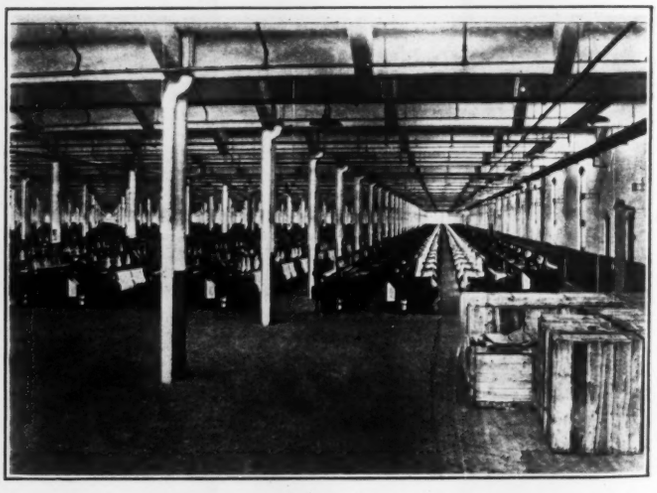

Fig. 12 Interior of Weave Shed of Cotton Mill

Cotton Mill

Fig. 11 shows the exterior of a cotton-mill weave shed in New England. The primary requirements are good light and economy of floor space. The latter involves proper column spacing. This plant is equipped with 3000 looms of various widths. The construction is reinforced-concrete columns and roof with brick walls. Fig. 12 shows the interior. It is a striking example of uniform lighting over a large area.

One of the most modern southern spinning mills is shown in Fig. 13. The building is plain, yet trim in every respect. Fig.

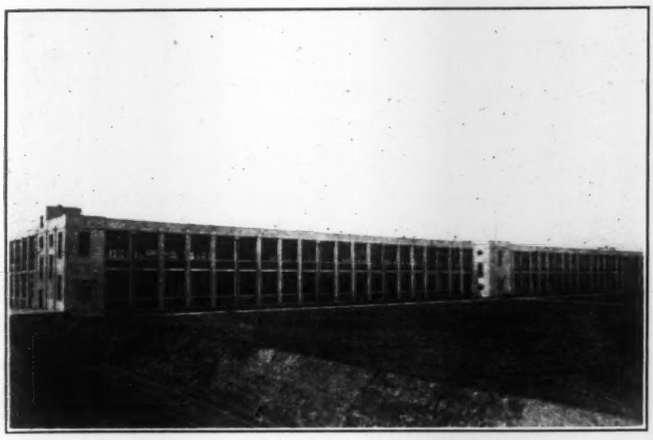

FIG. 13 COTTON-SPINNING MILL

14 shows the interior of the spinning room. The construction is a reinforced-concrete skeleton, with wood floors and roof and steel sash. The beams are floored with 4-in. plank, 1-in. intermediate floor, and $^{7}/_{8}$-in. maple top floor. The roof is of 3-in. plank with tar and gravel topping. The first floor, bearing on the ground, is a $4^{1}/_{2}$-in. concrete slab with a 2-in. creosoted plank bedded into 1 in. of sand and tar and a $^{7}/_{8}$-in. maple top floor. The cost of this building proved to be exceedingly low.

MAGAZINE PUBLISHING PLANT

Fig. 15 shows a magazine publishing plant on the outskirts of a middle-western city. It is laid out on one floor, lighted through sawtooth skylights above. As compared with former operations in a multi-story building in New York, the cost of production has been substantially reduced in the new plant.

Fig. 16 shows the opposite end of the building, with railroad siding, shipping and receiving platform, and power plant.

The press room is shown in Fig. 17. Air ducts provide forced ventilation where it is needed. Aisles are ample for access to machinery and for moving material. Uniform lighting is essential in this room where the highest grade of printing is done, much of it in several colors. Close temperature regulation at all seasons is required to make the ink flow properly.

Fig. 18 shows the bindery, where labor saving and increased production are accomplished by the correct arrangement of machinery and provision for handling material with the least possible effort.

A feature of this plant is the arrangement on one level, which makes mechanical conveying equipment unnecessary at the same time that it allows for future expansion of all departments without disturbing the present equipment.

The two-story section shown in the foreground of Fig. 15 is occupied by the plant office and circulation department. A similar

FIG. 15 MAGAZINE PUBLISHING PLANT

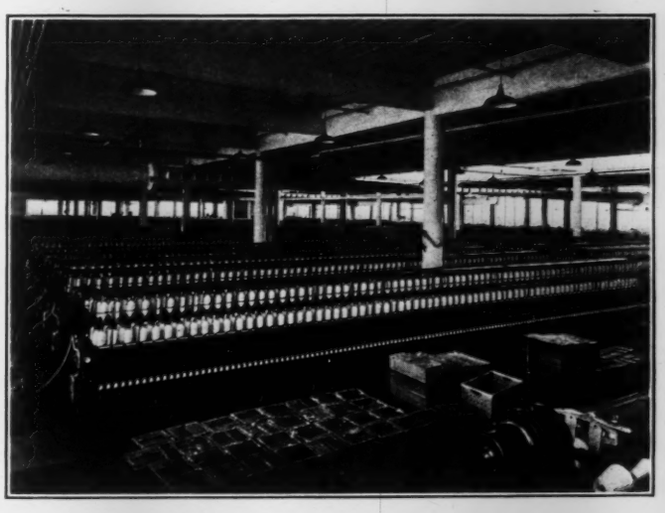

FIG. 14 INTERIOR OF COTTON-SPINNING MILL

section in the rear is taken up with composing room, miscellaneous equipment, and paper storage. The two-story sections are of reinforced concrete. The remainder of the building has a steel frame, with brick walls and gypsum roof. The floors are concrete, with Kalman finish.

NEWSPAPER PUBLISHING PLANTS

Modern newspaper publishing is essentially a manufacturing industry, with the same fundamental problems as any other manufacturing business. Metal and paper are handled by the hundreds of tons and printed papers are wrapped, mailed, and distributed by hundreds of thousands. The time element in production is vital. The entire product is consumed each day without the chance to balance production by carrying over a part. Labor is highly organized, working short hours and demanding high wages. The mechanical equipment is intricate and very expensive. The large investment needs to be protected by the best possible building design, to promote efficiency and reduce expenses.

Fig. 19 shows the arrangement of departments in the New York *Herald Tribune* plant. Editorial and composing rooms are on the upper floors; paper is delivered by a gravity chute to the basement and stored there. The mailing and delivery room is at street level, a requirement of first importance in this particular location. Presses are supported on the second floor, the third floor being an operating gallery around the presses. Everything moves continuously downward except paper from basement storage to the second floor. The alternative of presses below street level was rejected because of excavation expense and objectionable working conditions below ground.

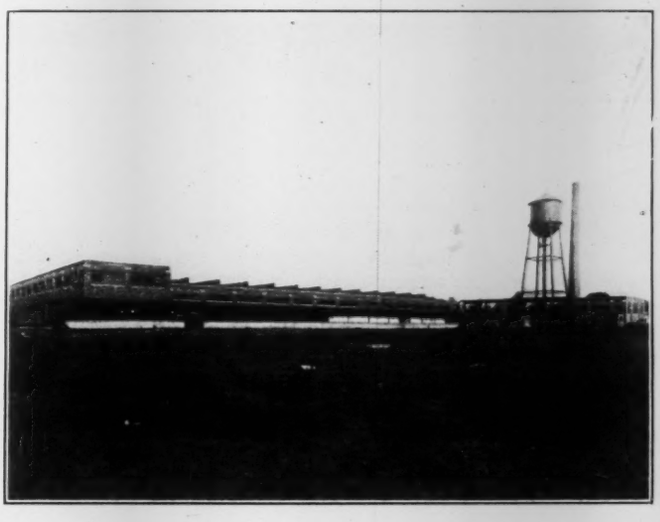

FIG. 16 MAGAZINE PUBLISHING PLANT SHOWING RAILROAD SIDING AND POWER HOUSE

FIG. 17 MAGAZINE PUBLISHING PLANT, PRESS ROOM

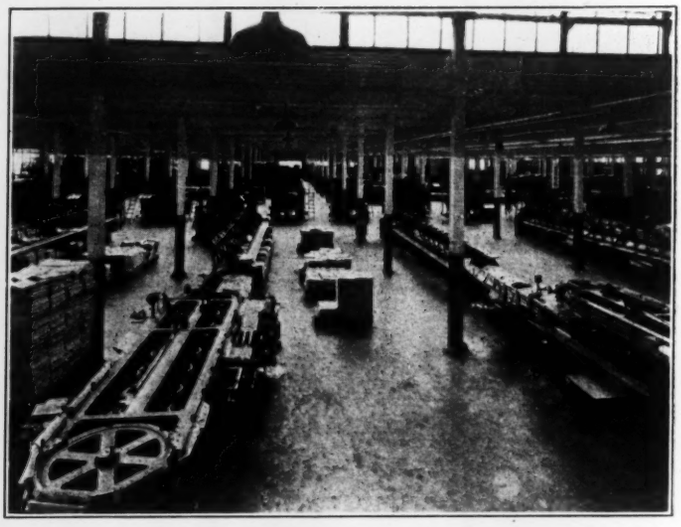

FIG. 18 MAGAZINE PUBLISHING PLANT, BINDERY

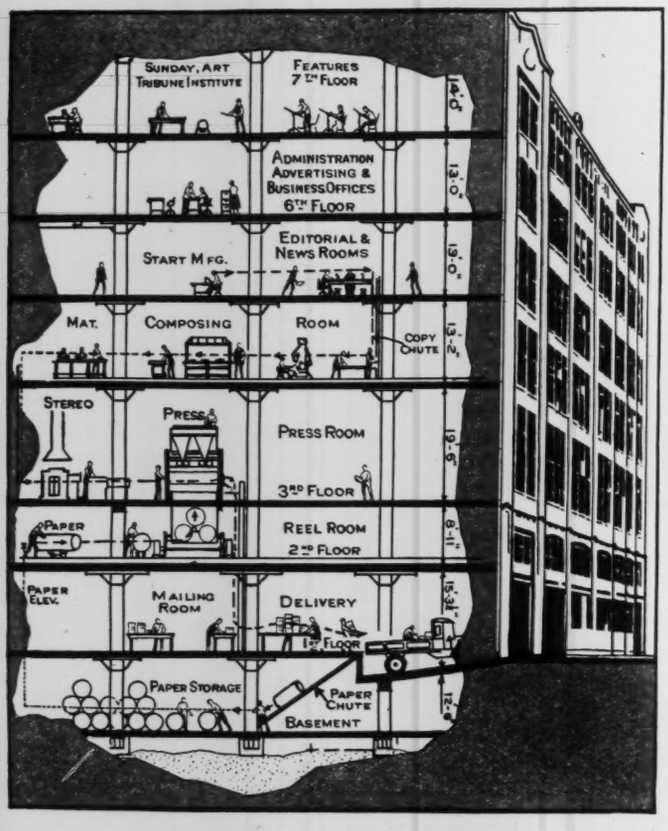

FIG. 19 NEWSPAPER PUBLISHING PLANT, ARRANGEMENT OF DEPARTMENTS

FIG. 20 NEWSPAPER PUBLISHING PLANT, PRESS ROOM

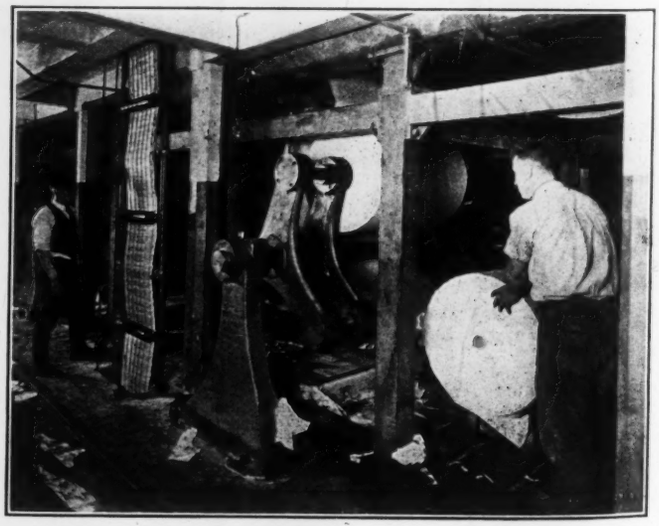

FIG. 21 NEWSPAPER PUBLISHING PLANT, PRESS SUPPORTS, REELS, AND PAPER CONVEYOR

The building is of reinforced-concrete flat-slab construction throughout. A structural problem was presented by mounting the heavy press vibrating load on the second floor. It was overcome by giving the floor slab sufficient mass to absorb vibration by inertia. This has worked out well in practice.

Fig. 20 shows the press room of the *Herald Tribune* and Fig. 21 the reels under the presses and the press supports on the second floor. A conveyor carrying printed papers from the press room to the mailing room below is shown at the left of the picture.

The latest development in metropolitan newspaper-plant design is illustrated by the Brooklyn plant of the New York *Daily News*. The owners announce that it is the first of a series of auxiliary printing establishments called for by the growth of the *News*. It is designed to print the Brooklyn and Long Island circulation. The alternative would have been to enlarge the Manhattan plant. This proved to be more expensive and less efficient than to make

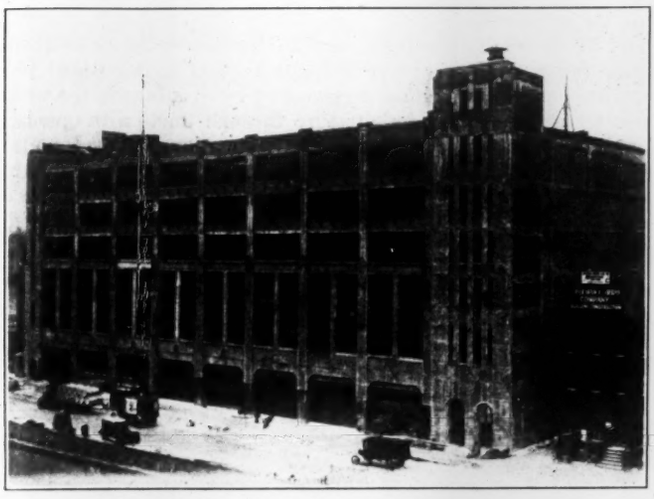

FIG. 22 NEWSPAPER PUBLISHING PLANT, EXTERIOR BEFORE COMPLETION

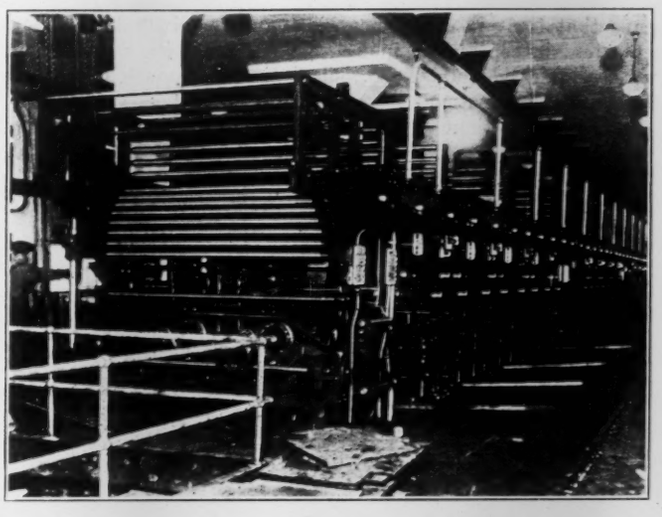

FIG. 23 NEWSPAPER PUBLISHING PLANT, PRESS ROOM: LOWER VIEW

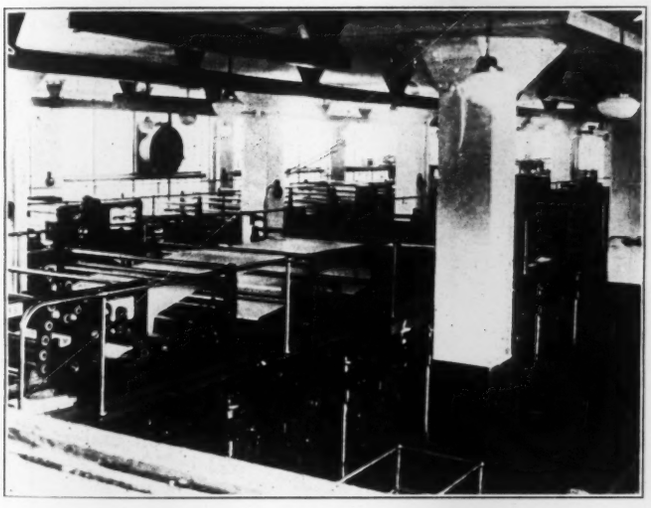

FIG. 24 NEWSPAPER PUBLISHING PLANT, PRESS ROOM, UPPER VIEW

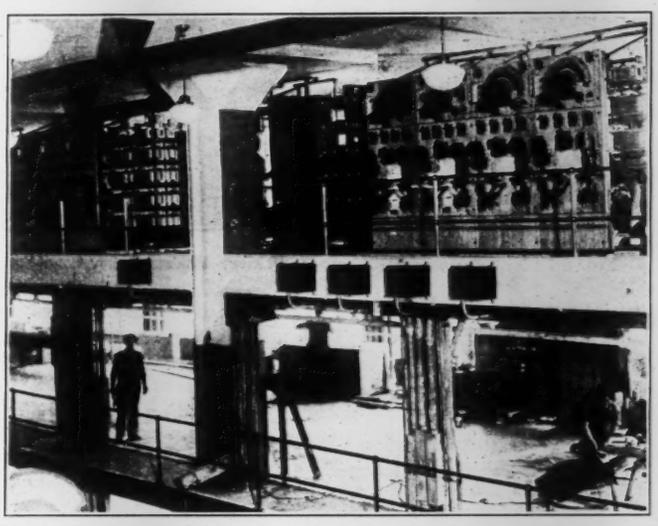

FIG. 25 NEWSPAPER PUBLISHING PLANT, PRESS ROOM, CONTROL GALLERY

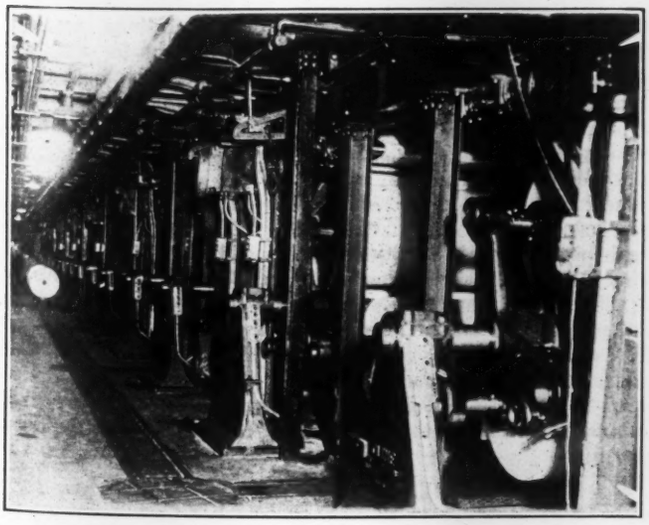

FIG. 26 NEWSPAPER PUBLISHING PLANT, PRESS SUPPORTS, REELS, AND INDUSTRIAL TRACK

FIG. 27 NEWSPAPER PUBLISHING PLANT, STEREOTYPING FOUNDRY

extra matrices in Manhattan and send them to Brooklyn where the process would begin with casting plates for the presses. This means that the editorial, composing, and matrix-making departments remain centralized in Manhattan, while the papier-maché matrices, quickly made, light and easy to carry, are sent to the branch plant for casting.

The Brooklyn plant of the *News*, therefore, is strictly a manu-

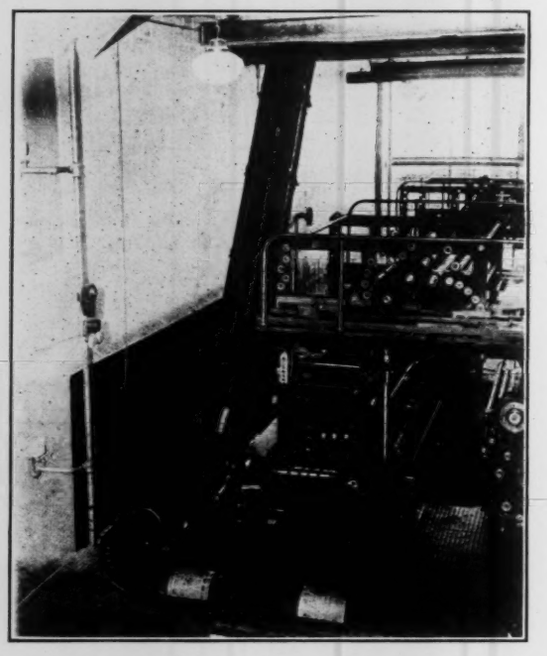

FIG. 28 NEWSPAPER PUBLISHING PLANT, PLATE CONVEYOR

facturing plant. Its design was governed by considerations of quality and cost of production, with adequate provision for paper storage and a location selected purely on the basis of easy, quick distribution.

Fig. 22 shows the exterior of the *News* building shortly before completion. It is a flat-slab, reinforced-concrete structure of six stories and basement, with provision for three additional floors in the future. The total floor area is about 160,000 sq. ft. The basement is devoted to a heating plant, electric substation, coal bins, well and pump room, ink storage, and paper storage. The first floor is all mailing room and space for trucks to back in clear of the sidewalk. This space is sufficient for ten trucks at the delivery platform and two at the paper-receiving platform. It is enclosed by the large rolling doors shown clearly in the picture. Presses are supported on the second floor, the same as in the *Herald Tribune* plant already described. The third floor is a steel operating platform supported by the second floor. The fourth floor is occupied almost entirely by paper storage, except space on one side where the stereotype foundry is installed. Space over the stereotype foundry on the fifth floor is occupied by ventilating equipment. The remainder of the fifth and six floors is paper storage.

Figs. 23 and 24 are press-room views. There are two rows of presses, each having 18 units, already installed and space left for another row. Fig. 25 shows the control gallery in the press room, with the electrical controllers and selector panels that shift the control push-button stations on the press units to the right drive in different combinations. Fig. 26 shows the press supports on second floor, reels under presses, and the industrial track system for delivering paper to reels. The second floor is designed to carry a loading of 900 lb. per sq. ft., with mass sufficient to absorb press vibration by inertia.

Fig. 27 shows the stereotyping foundry above the press room. Ventilating hatches in the ceiling above each machine lead to a compartment from which air is exhausted by disk fans in the walls. Fresh air is supplied under pressure through ducts with openings at intervals along the rear walls near the floor level of the foundry. New plates are carried by a mechanical conveyor from the foundry to the press room below. The conveyor runs the full length of the press room and delivers plates to all units. The same conveyor continues back to the foundry to return used plates for remelting. The return end is shown in Fig. 28, rising from press room to foundry level.

Printed papers are carried from the press room to the mailing room, two floors below, on specially designed vertical conveyors divided into units which are set at intervals convenient to the

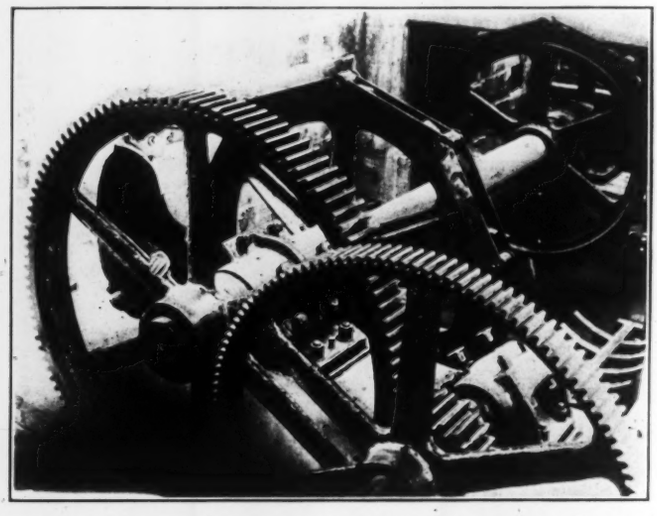

FIG. 29 NEWSPAPER PUBLISHING PLANT, HEAD MECHANISM OF PAPER CONVEYOR

folders. Carriers mounted on a continuous chain and sprocket mechanism deliver papers in batches of 300 to 400 to horizontal-belt conveyors which deposit them on wrapping tables.

The storage of such a large amount of paper (5500 tons, estimated as three weeks' supply) on several floors requires exceptional handling facilities. One of the most serious items of expense in a large newspaper plant, anyway, is paper handling. Ordinary freight-elevator service is too slow and laborious for the *News* requirements, which are three rolls of paper per minute, each roll weighing 1500 lb. This paper has to be received from trucks at the street level and delivered to any of the upper floors at will, taken from one floor to another, or brought back from storage to trucks for delivery to another plant.

The requirements named above have been met by a continuous chain conveyor serving all floors and built heavy enough to be lengthened for three additional floors whenever the building is enlarged. The head mechanism of this conveyor is shown in Fig. 29. One of the cradles that carries a roll of paper is shown at the right. At each floor there are supporting fingers that can be set either for loading or unloading. The conveyor is reversible, so it can load or unload from either side on any floor. Paper stored in the basement is sent down from the truck by a gravity chute and carried to the press, when needed, by the conveyor.

Diesel Traction for Railroads

General Considerations—Advantages of "Dieselizing" Certain Railroad Services—Operating, Design, and Construction Problems—Weights and Speeds—Cooling, Transmission, and Vibration Problems—Maintenance—Fuel Consumption—General Conclusions Favoring Adoption of Diesel-Electric Traction

By WILLIAM ARTHUR,[1] CAMDEN, N. J.

IT IS ABOUT 100 years ago that the first crude forms of steam locomotives were developed, and no serious rival appeared until electric traction made its appearance some 30 to 40 years ago. Development in both steam and electric branches has continued, the one acting as a stimulus to the other. High efficiencies are now possible with either modern steam locomotives or electric locomotives, or with multiple-unit rail cars under any of the well-known and well-tried systems.

Subsequently the gas-electric car made its appearance on our railroads, and has met with considerable favor as an auxiliary on branch or feeder lines. Nevertheless certain inherent drawbacks in gas-electric traction, notably the relatively high cost of gasoline in the larger units, has led railroad operating officials to look for other traction means.

More recently the further development of the Diesel oil engine has

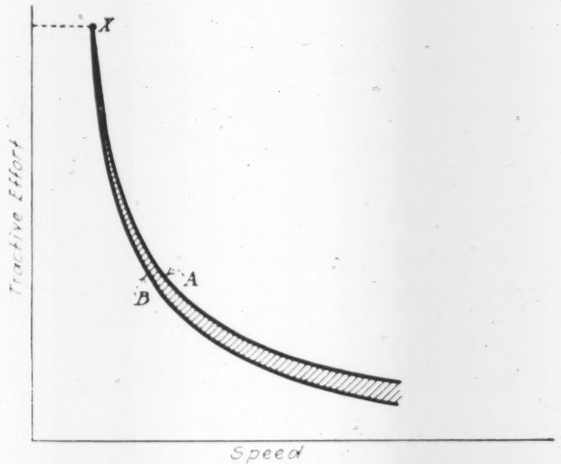

FIG. 1 SPEED VS. TRACTIVE EFFORT, DIESEL AND ELECTRIC LOCOMOTIVES

introduced new possibilities and many railroads are utilizing this latest form of traction, employing Diesel engines carried on the traction unit itself; more than fifty Diesel locomotives and rail cars are already either in use or under construction in different parts of the world, so that the "Dieselizing" of railroads may be said to be already started.

In the most usual case the traction unit consists of a Diesel engine driving an electric generator, which in turn is connected to electric motors carried on the trucks in a manner similar to that employed in ordinary electric rail cars or locomotives. In other cases the drive is through gears. The advantages of "Dieselizing" certain railroad services may be summarized as under:

(*a*) *Elimination of Smoke and Noise.* The Diesel unit shares these advantages with its electric rival, and with good design operates smokelessly and without undue noise under all working conditions. It therefore possesses advantages for terminal work and for use in switching services in locations surrounded by valuable residential property, similar to those enjoyed by electric traction units.

(*b*) *Elimination of Terminal Delays.* The need of elaborate coaling and water stations, ashpits, etc., together with the apparatus and services dependent upon these features, is eliminated. The Diesel unit is ready immediately for service, and no time is lost in getting up steam or in the various other "hostling" services so necessary with steam locomotives. In this again it is similar to the electric locomotive.

(*c*) *Economy of Installation.* When compared with straight electric traction the need of expensive power stations, substations, transmission lines, and overhead or third-rail contact systems with their accompanying costs and dangers to life, is done away with. The Diesel unit is self-contained; it carries its power station on its own back. The economics in each case have to be separately studied, but the advantages of the new form of traction, particularly for branch lines and for switching services, are already well demonstrated in this country and in Europe, and larger units are now being introduced into main-line road services.

(*d*) *Fuel Economy.* The fuel cost, which of course with all forms of traction is one of the major items of expense, is much reduced and works out in round figures for the same service as follows:

Steam locomotive	Electric locomotive	Diesel locomotive
100 per cent	50 per cent	20 per cent

and on the basis of thermal efficiency at rail:

Steam locomotive	Electric locomotive with modern power station and transmission systems	Diesel-electric locomotive
5–9 per cent	10–12 per cent	22–25 per cent

The problems that arise in connection with Diesel traction apparatus come under two general headings:

a Operating problems
b Problems of design and general arrangement.

Of course these matters are closely related, in fact in practice they cannot be separated; nevertheless for the sake of clarity the author will herein discuss them separately.

OPERATING PROBLEMS

Diesel traction differs from electric traction in that with the latter where energy is taken from an overhead wire or third rail, there is usually a large power plant to draw from and from which, for short periods, large overload demands can be made, whereas with the former there is a definite limit fixed by the horsepower capacity of the Diesel unit itself. In practice, a Diesel engine is given a certain normal continuous horsepower rating at normal speed and is capable of overloads of only 20 to 25 per cent for short intervals. It shares this limitation with the ordinary steam locomotive, although recent developments in Diesel design are tending to remove it and extend the overload capacity to a point thought impossible only a few years ago.

On the other hand, the Diesel unit possesses the ability (when combined with an electric transmission system) of translating its full rated horsepower into either high tractive effort at low speed, or into the equivalent high speed at lower tractive effort. In this, like the steam locomotive, it is markedly superior to the electric locomotive operating under a fixed voltage in overhead wire or third rail. This can be better understood by referring to Fig. 1. Here *A* represents the constant-horsepower line which can be closely followed with either a steam or a Diesel locomotive, and *B* represents the condition which obtains with practically all forms of electric locomotives. For instance, if the electric locomotive is designed and geared to deliver its full rated continuous horsepower output at the tractive effort and speed corresponding to the point *X*, then at higher speeds there will be a marked falling off in tractive effort

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Be sure to bring this copy with you to the Spring Meeting

which, in certain instances, will amount to as much as 30 per cent. Electric locomotives must therefore be designed very closely and to some extent limited to the particular service for which they are intended. This is not the case with Diesel-electric locomotives. This inherent flexibility of the latter type is a valuable asset in railroad operation, where in practice tonnage ratings are determined not merely by considerations of ruling grade but by schedule requirements as well. The shaded area between curves *A* and *B* can, in the case of the Diesel-electric locomotive, be entirely filled in by reason of the flexibility of the voltage control which is possible.

Summarizing this portion of the paper, it may therefore be said that the Diesel-electric locomotive possesses most of the advantages of both steam and electric locomotives, but without some of the disabilities of each.

Problems of Design and Construction

Diesel-Engine Types. The engineer interested in the design of Diesel traction apparatus is confronted with a multiplicity of types of Diesel engine from which to choose—2 or 4 cycle, single or double acting, with air injection or with solid injection, with 4, 6, 8, or 12 cylinders arranged in line or in V-construction, with vertical, horizontal, or radial types. Each possesses certain advantages and disadvantages in specific situations; the whole art is developing rapidly, and interesting and significant experiments are under way in the U. S. A., in Sweden, England, Russia, Germany, Switzerland, France, Italy, Africa, and elsewhere.

The author has recently completed a tour lasting more than a year, made expressly for the purpose of studying the outstanding developments in the U. S. A. and in Europe. He was given opportunity to visit practically every important railroad or manufacturing establishment interested in this kind of development, and the present paper embodies some of his findings and conclusions based upon this study.

In considering Diesel-engine types the first choice has to be made between:

a 2-cycle types versus 4-cycle types
b Air-injection versus solid-injection types.

The chief advantage of the 4-cycle types over the 2-cycle is that no scavenging pump is required. This pump is a relatively large and costly piece of apparatus, requiring considerable power, usually about 6 per cent, to drive it. The chief advantage claimed for 2-cycle operation is that a reduction in total weight and cost takes place. In theory this should be so, but in practice in a multiplicity of designs studied by the author this advantage does not seem to have been realized, one of the chief reasons being that due to the need for effective scavenging with 2-cycle types, lower piston speeds are necessary, and total weights and costs increase in consequence.

With air injection the fuel oil is blown into the combustion chamber by air at high pressure, usually at about 850 lb. per sq. in. This requires an air compressor of special design, coolers, pumps, etc., and other expensive apparatus. Moreover, to drive the compressor takes 10 per cent of the total power. With solid injection, however, the oil is squirted directly into the cylinder at each stroke, requiring for this purpose only a small simple and relatively inexpensive oil pump. Air-injection types are therefore inherently more complicated and expensive than solid-injection types.

The chief disadvantage of solid injection is that if the quality of the fuel oil is poor, and contains, for instance, much solid matter, such as asphalt, ashes, etc., trouble may result from the clogging up of the small pinholes through which the oil is injected. Unequal explosions then take place and dangerous pressures become a possibility. This is an important consideration in marine work, for a ship cannot always control the source of supply of its oil and may have to use what it can get. This factor is, however, not of equal importance in land traction work, where usually oil can be obtained of a uniform quality. All things considered, airless- or solid-injection types are to be preferred for traction purposes.

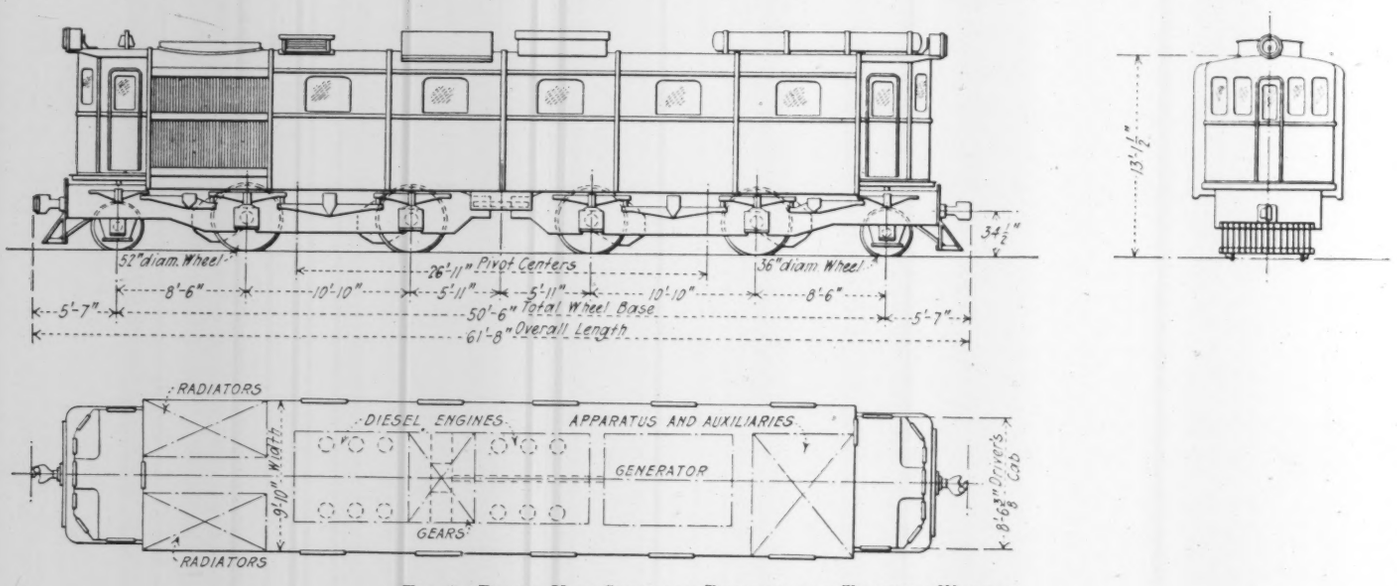

FIG. 2 DIESEL UNIT SPECIALLY DESIGNED FOR TRACTION WORK

Weight and Speeds

Diesel locomotives and rail cars in the past have been notable for their relatively large weight and consequent high cost, amounting in certain instances to from two to three times that of an equivalent steam locomotive. This condition has arisen from two general causes. First because no Diesel engines had heretofore been developed specially for traction purposes and the locomotive designer was forced to take some existing relatively low-speed and heavy unit designed originally for submarine or stationary purposes and utilize it for traction. Similarly for the electrical requirements, generator, motors, etc., with consequent increase in the weight of all contributary apparatus such as mechanical structure, trucks, frames, etc. Secondly, it has been thought sufficient to simply take a Diesel engine, a generator, motors, and control, together with the necessary fuel and other storage tanks and apparatus, locate and connect them together in a suitable mechanical frame and call the result a Diesel locomotive. In consequence, weights have reached as high as 400 lb. per b.hp. as compared with less than half of this amount for an equivalent steam locomotive. Even in one of the best and most successful recent American designs, employing a 600-hp. Diesel engine, the total weight is 100 tons on drivers, giving 333 lb. per b.hp., or only 6 hp. per ton on drivers.

In certain instances, where the service for which the locomotive is designed is one wherein large tractive effort at relatively low operating speeds obtains, i.e., in switching services and where weight is necessary for adhesion purposes, these high weights per horsepower may be justified; but even in these cases it is doubtful if a substantial decrease in weight would not increase the "liveliness"

of the unit—a characteristic most useful and necessary in such services.

The author has just completed designs for a number of Diesel-electric locomotives for use on American railroads in which he has endeavored to avoid these deficiencies and limitations; and by utilizing a Diesel unit specially designed for traction work, by close regard to the interdependence of parts, i.e., by using the Diesel frame in part to support and supplement the main locomotive frame by designing and dovetailing the Diesel engine and its generator into a composite unit, thereby shortening the length of the whole locomotive in short, by coördinating and combining the best American and European practice, we obtain the unit as illustrated in Fig. 2.

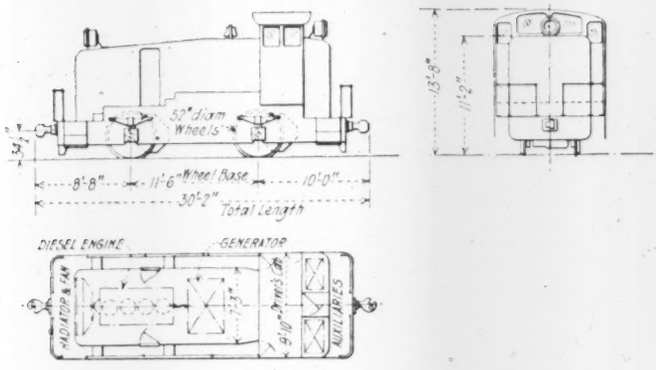

FIG. 3 SMALLER DIESEL UNIT DESIGNED FOR SWITCHING SERVICE

The general characteristics of this locomotive are:

Track gage	4 ft. 8½ in.
Diameter of driving wheels	52 in.
Diameter of guiding wheels	36 in.
Rigid wheelbase	10 ft. 10 in.
Total wheelbase	50 ft. 6 in.
Overall length	61 ft. 8 in.
Output of Diesel motor, continuous	1600 hp.
Output of Diesel motor, maximum	1850 hp.
Weight of Diesel and electric equipment, including 9000 lb. of fuel and 2200 lb. of cooling water	164,000 lb.
Weight of mechanical part	132,000 lb.
Total weight	296,000 lb. = 148 tons
Weight on drivers	220,000 lb. = 110 tons
Gear ratio	23:77
Tractive effort at wheel rim, continuous	13,400 lb.
Corresponding locomotive speed	34.6 m.p.h.
Tractive effort at wheel rim, 1 hr.	17,800 lb.
Corresponding locomotive speed	26.0 m.p.h.
Tractive effort at wheel rim, maximum	35,000 lb.
Maximum locomotive speed	56.0 m.p.h.

It will be noted that 14.5 b.hp. per ton of weight on drivers has been obtained. The weight on drivers per continuous b.hp. developed is 138 lb., and for the total locomotive, 185 lb. per b.hp.

The locomotive is designed for road service and for use in either freight or passenger duty by suitable changes to the gear ratio.

A smaller unit designed for switching services is shown in Fig. 3. The general characteristics of this locomotive are:

Track gage	4 ft. 8½ in.
Diameter of driving wheels	52 in.

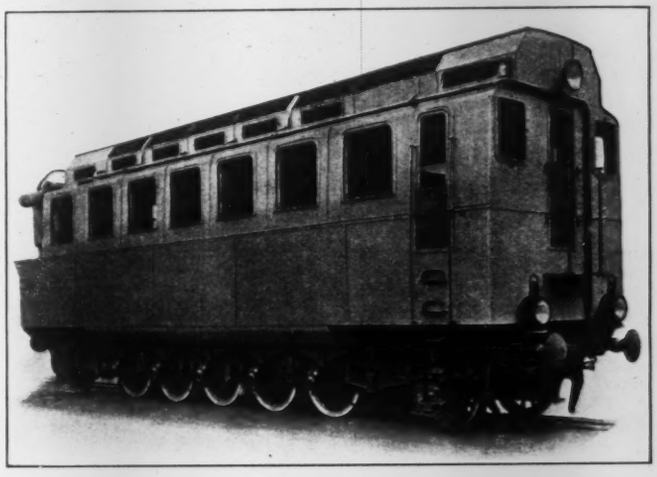

FIG. 4 A MODERN EUROPEAN DIESEL LOCOMOTIVE

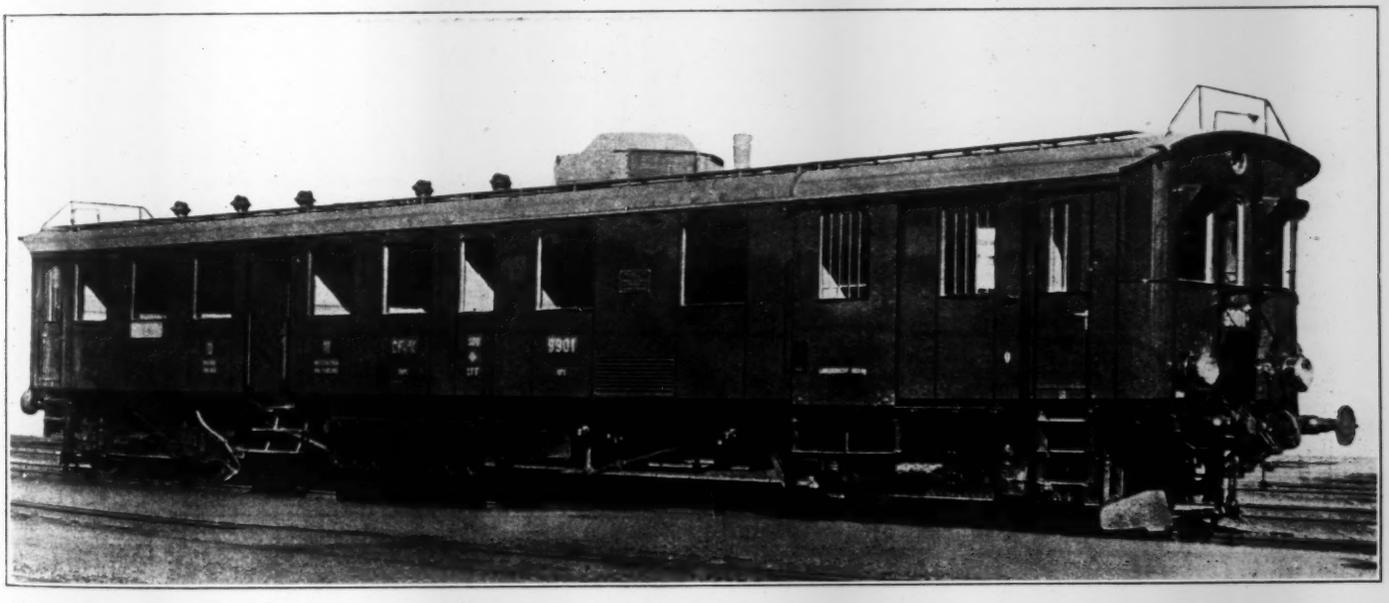

FIG. 5 ONE TYPE OF EUROPEAN RAIL CAR

Wheelbase	11 ft. 6 in.
Overall length	30 ft. 2 in.
Output of Diesel motor, continuous	400 hp.
Output of Diesel motor, maximum	500 hp.
Weight of Diesel and electric equipment, including 1320 lb. of fuel and 800 lb. of cooling water	54,000 lb. = 27.0 tons
Weight of the mechanical part	55,000 lb. = 27.5 tons
Total weight	109,000 lb. = 54.5 tons
Gear ratio	16:84
Tractive effort at wheel rim, continuous	9000 lb.
Corresponding locomotive speed	12.4 m.p.h.
Tractive effort at wheel rim, 1 hr.	13,200 lb.
Corresponding locomotive speed	8.4 m.p.h.
Tractive effort at wheel rim, maximum	27,500 lb.
Maximum locomotive speed	28.0 m.p.h.

An intermediate unit suitable for terminal switching and lighter road services has the following general characteristics.

Output of Diesel motor, continuous	800 hp.
Output of Diesel motor, 1 hr.	1000 hp.
Weight of Diesel and electric equipment, including 4500 lb. of fuel and 1320 lb. of cooling water	93,000 lb.
Weight of mechanical part	85,000 lb.
Total weight	178,000 lb.

Weight on drivers	119,000 lb.
Tractive effort at wheel rim, continuous	6700 lb.
Corresponding locomotive speed	34.6 m.p.h.
Tractive effort at wheel rim, 1 hr	8900 lb.
Corresponding locomotive speed	32 m.p.h.
Tractive effort at wheel rim, maximum	17,500 lb.
Maximum locomotive speed	56 m.p.h.

Fig. 4 is representative of modern European Diesel locomotive practice and Figs. 5, 6, and 7 show two types of European rail car.

The Diesel engine utilized in the rail car shown in Fig. 5 is illustrated in Fig. 8.

Leaving now problems of general design for a more detailed consideration of the subject, certain new problems arise in Diesel traction apparatus which have to be very carefully studied in each case. These are:

1 The cooling problem—that is to say, the dissipation of the waste heat generated in Diesel motors

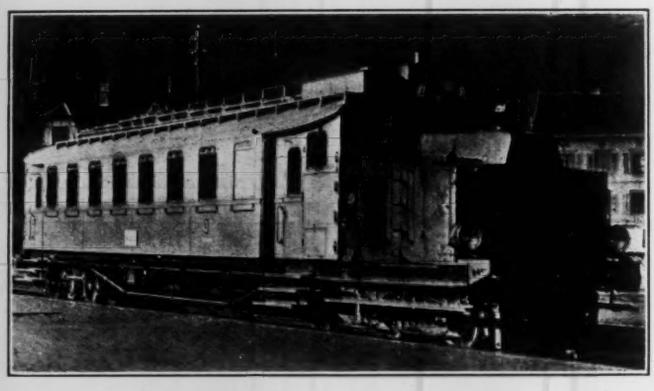

FIG. 6 ANOTHER TYPE OF EUROPEAN RAIL CAR

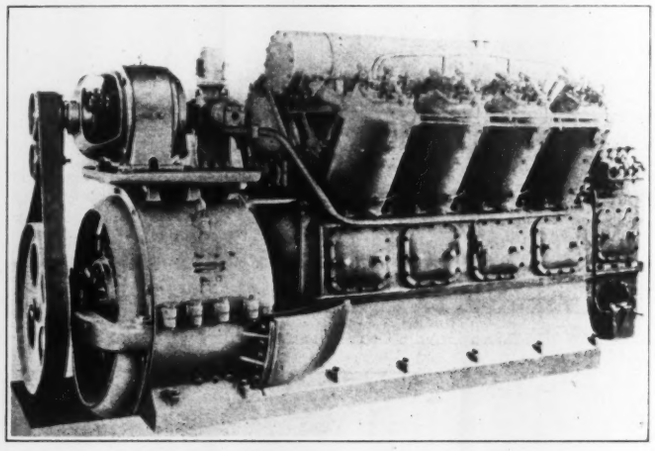

FIG. 8 DIESEL ENGINE USED IN RAIL CAR SHOWN IN FIG. 5

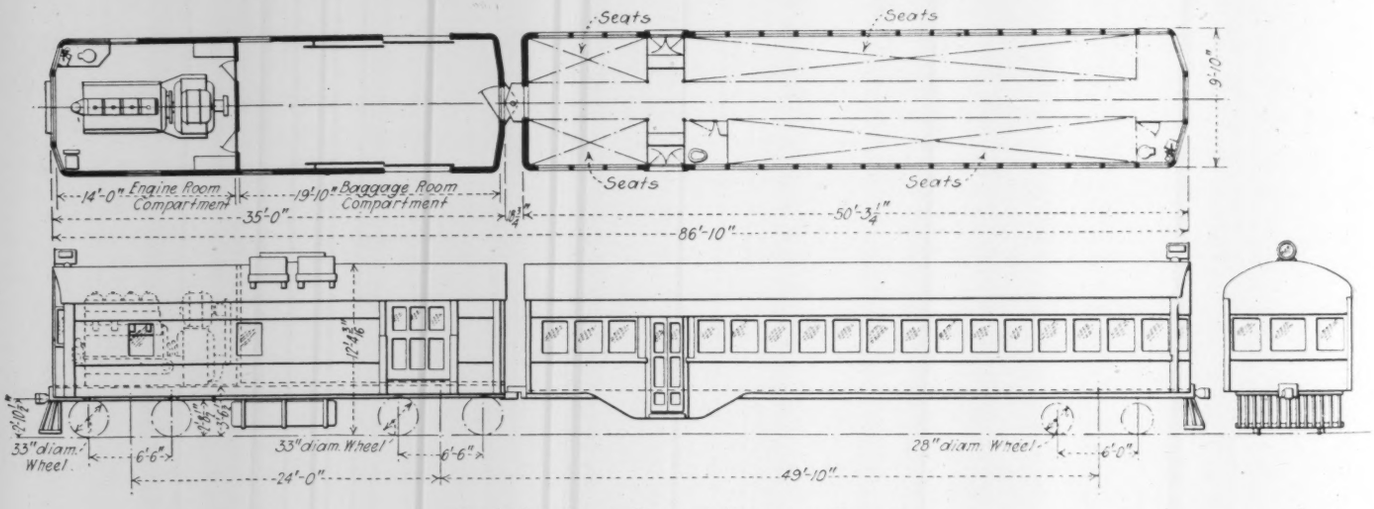

FIG. 7 A THIRD TYPE OF EUROPEAN RAIL CAR

Output of Diesel motor, continuous	400 hp.
Output of Diesel motor, 30 min.	500 hp.
Weight of Diesel and electric equipment	39,545 lb.
Gear ratio	17:66
Tractive effort at wheel rim, continuous	3700 lb.
Corresponding car speed	30 m.p.h.
Tractive effort at wheel rim, 1 hr	5500 lb.
Corresponding car speed	20 m.p.h.
Tractive effort at wheel rim, maximum	11,000 lb.
Maximum car speed	50 m.p.h.

2 The method of transmission of power from the Diesel prime mover to the wheels; and

3 The vibration problem.

THE COOLING PROBLEM

The importance of this can be at once appreciated when it is remembered that in the operation of a Diesel motor high pressures and high temperatures are generated, pistons, cylinder walls, exhaust valves, etc., have to be artificially cooled, and that in the final result approximately only one-third of the energy in the fuel oil reappears as useful work at the main shaft, another third disappears with the exhaust gases, and the final third reappears as heat which has to be gotten rid of in some way by radiation.

In an average case approximately 2500 B.t.u. have to be dissipated per hour per b.hp., and with a locomotive of, say, 1000 hp. the amount of energy to be dissipated by the cooling system constitutes a real problem.

There is great diversity of opinion as to the best way to solve the cooling problem, also as to whether to use natural draft, fans, or forced draft; whether to use the automobile or tubular type of radiator; whether to use oil or water or no cooling at all for the pistons; whether to drive the fans directly or by individual electric drive or by an exhaust-gas turbine through gears; how to best arrange the apparatus on the locomotive or rail car, having regard to the need for repairs or removal of the main Diesel unit, etc. All of these are questions which have to be settled in each individual case, and depend so much upon the horsepower involved, schedule speeds, local weather and temperature conditions to be coped with, etc., that is not possible to give a general answer to all of them.

There is, moreover, a relation between the area of the radiator surface and that of the combustion surface. A ratio of from between 30 and 40 to 1 seems to meet most cases. As stated, however, these matters have to be studied and settled in each individual case. Enough has been said to indicate the general nature of the cooling problem.

THE TRANSMISSION PROBLEM

The Diesel motor is a prime mover which cannot start itself. It has to be started by auxiliary means and run up to a certain speed, after which it will maintain itself in motion. As already pointed out, a steam locomotive is one of the most flexible pieces of apparatus

ever devised for traction purposes. Subject only to the limits of its steaming capacity, it can translate its horsepower into either tractive effort or speed to any reasonable degree desired, merely by varying its cut-off and mean effective pressure. It can exert its maximum tractive effort right at the start and when it is most needed, and then later, when the train is once accelerated, can utilize its full horsepower in the form of reduced tractive effort and increased speed. To a lesser degree, as explained, the electric locomotive can do the same, but the direct-connected straight Diesel locomotive cannot. It is necessary to interpose between the Diesel main shaft and the wheels some form of transmission device to make up for this inherent deficiency of the Diesel motor. In this connection it may be mentioned that much developmental work is in progress in Europe and elsewhere looking to extend the flexibility of Diesels, and that substantial gains have already been made.

There are five general methods of transmitting the torque from the prime mover to the wheels. These are:

1 By direct mechanical drive through a clutch or clutches
2 Through mechanical gearing
3 Through hydraulic gearing
4 By mixed systems in which either steam or compressed air is used to augment the flexibility of the Diesel unit at starting.
5 By electric transmission.

Direct Drive is not practical because of the reasons above referred to, although several attempts have been made to utilize it. The Sulzer Company built a locomotive with direct drive but it was not successful; others have tried it and failed, and we can safely dismiss further consideration of this method.

Mechanical Gearing is practical, but at present is limited to smaller apparatus, say, below 300 hp., because of the very heavy stresses on the gear teeth, particularly when a train is suddenly started; and also because of the difficulty in designing a clutch which will cope successfully with these large forces. Possibly progress will take place in this direction in the future, for many interesting and courageous experiments are under way in the United States, in France, Switzerland, Germany, and elsewhere. It would be going too far afield to enumerate these.

Hydraulic Gearing offers possibilities and usually consists of a number of primary pumps or pistons delivering oil under pressure to a series of secondary pumps or pistons, the stroke or cubical dimensions of which can be varied at will, thereby permitting an infinite variety of speed changes. There are many types, of which the Lentz, the Hele-Shaw, and the Schneider are the best known. The Schneider gear, developed by the Swiss Locomotive Works at Winterthur, is one of the best. A 500-hp. unit weighing 11 tons has been built and has an efficiency of 85 per cent. Bigger sizes up to 1600 hp. are, it is claimed, practicable. Such apparatus is, however, quite heavy and expensive, and has not yet been tried out on any large scale. There are unsolved problems with regard to free coasting, reversing, etc. Developmental work is still in progress. The fact that so many new types and variations of the same basic idea are being produced, none of which gets very far, is indication that the problem is not yet completely solved.

Special Forms of Mixed Transmission Systems have been developed and are being tried out. These are:

a The Still System in England, whereby steam (from a steam boiler) is used to supplement the Diesel deficiencies at starting and during acceleration
b The Zarlatti System and the Cristiani System (both in Italy), wherein compressed air and compressed steam are used in separate cylinders as a steam locomotive and so supplement the Diesel prime mover when necessary.

Electric Transmission in the present state of development is, in the author's opinion, and for all except the smallest sizes, the only one we can now safely adopt. General opinion in Europe largely concurs in this. Its use simplifies the starting and removes the need of reversing arrangements on the Diesel prime mover, all reversing, of course, being executed in the most simple manner by changes to the motor leads. Electric transmission permits the Diesel motor to be run at its most efficient speed at all times; all train-speed control being taken care of by voltage variations on the generator, and where necessary by series-paralleling and field-control arrangements on the motors. Electric generators and motors are standardized pieces of apparatus of high efficiency and great ruggedness. We know exactly what they can be counted on to do under any ordinary circumstances. Exact figures are not available, but from such studies as he has made, the author believes that ultimately very little difference in cost or weight will be found between electric and hydraulic methods for the larger types, although there may be cases where in smaller units mechanical gears will show a lower first cost and weight. These alternatives should of course be carefully studied in each case as it arises. For reasons given, however, it is felt that on the whole electric transmission is the best solution for all but the small sizes, and even in these it should be seriously considered because of the many other incidental and fundamental advantages which its use makes possible. The advantages of electric transmission are becoming more and more appreciated, and it would not be surprising to see it universally adopted in the next few years for all Diesel traction apparatus excepting the very small units.

The Vibration Problem

Diesel traction is a relatively new branch of engineering and the combination of high Diesel speeds and high unit pressures leads to new and to a large degree unforeseen results.

One of the definite facts that has been developed by recent practice is that the problem of vibration and critical speeds becomes of great importance. An accelerating and rotating mass—as, for instance, a Diesel motor which is being speeded up—passes through a series of critical speeds, at which points the value of the forces brought into play may be many times greater than normal. The critical speeds may be below the normal speed and therefore the unit must always pass through these on getting up to full speed. These values change as the apparatus attached to the main shaft is changed; and a Diesel may operate perfectly alone, and yet when connected, say, to a generator on the same shaft (as in Diesel-electric locomotives), may with faulty design develop undesirable and unpleasant vibrations.

The matter of vibration becomes of even more importance when Diesel rail cars are considered, for with faulty design the resulting noise and unpleasantness make the car unpopular with the public.

Provision Against Freezing

Arrangements are usually made so that in winter if the unit stands for more than a few minutes, the radiators automatically empty themselves into a cooling-water storage reservoir located inside the vehicle. A small electric heating element is sometimes provided to be used in emergencies and where the stop in cold weather is likely to be of long duration.

The storage tank should of course be located at a point below the Diesel cylinders, in order that there may be no risk of cracking the cylinder walls. The exact arrangement must be studied in each individual case, for in certain instances an electric heating unit might put too much strain on the battery, making it necessary to provide other means—possibly a small oil-fired heating unit.

The Maintenance of Diesel Apparatus

The art is still too new for there to be much real information available as to cost of maintenance; also types vary so much that such information would in any event be of little value. This much can be said, however: Any of the well-known makes of Diesels, either in this country or in Europe, operate quite reliably. This is true of either 2-cycle or 4-cycle types or those with air or airless injection. Some wear and tear of course takes place. Exhaust valves are subject to high temperatures and pressures and need inspection periodically. About two weeks a year should be allowed for general overhaul.

The Starting of Diesel Motors

In considering the problem of starting Diesel motors for traction apparatus, one should differentiate between locomotives of large size and locomotives of small size or rail cars; the problem also changes, depending upon whether or not the service is one requiring frequent stops.

There is a great difference of opinion on this whole matter. Some manufacturers prefer air starting, using all of the cylinders for this purpose; others use only a few of the cylinders or possibly one cylinder; still others use a separate air motor geared to the main motor, the advantages of the latter arrangement being that the cylinder walls are not cooled by the air at the moment of starting and a quicker start can thereby be made.

The problem of starting is tied in with that of the method of transmission adopted, for with apparatus of moderate size, if, as is expected, electric transmission is to be used, then in certain cases it is simpler and better to utilize the generator as a starting motor and provide a battery for this purpose. For large units, however, the weight and cost of a battery may be prohibitive.

In locomotive services involving long runs and few stops it would not be necessary to start the Diesel frequently. In fact, it is conceivable that one might start it and keep it running for most of the day in such services.

Diesel Piston Speeds

Every manufacturer is of course striving to reduce his weight per brake horsepower, particularly for traction or submarine work. One of the chief ways of reducing weight is to increase the piston speeds. The limit at present seems to be about 1250 ft. per min.

Revolutions per minute of Diesel apparatus vary very much. Most of the light-weight, high-speed Diesels hitherto built were designed originally for submarine purposes and usually ran at about 450 r.p.m. The recent tendency is, however, to increase speeds very much, and those of from 600 to 1000 r.p.m. are not now uncommon.

Pressures Used in Diesel Apparatus—Fuel Consumption

In general, the compression in Diesel cylinders varies from 500 to 600 lb. in European practice, although in the United States the pressures are somewhat lower.

The chief problem in the operation of a Diesel motor from the point of view of fuel consumption is of course to insure that complete combustion takes place. With air-injection types this is taken care of in the most thorough manner by completely breaking up the oil by means of the high-pressure injection air.

In general, it may be safely said that today there is not much difference in fuel consumption between air-injection and solid-injection types, and a fuel consumption of 0.4 lb. of fuel oil per b.hp-hr. is obtainable under ordinary conditions with either type.

General Conclusions

The author believes that the next few years will show a marked increase in the number of Diesel applications to specific railroad problems, as, for instance, where bad water conditions obtain, and particularly to switching service, to branch lines, and to a lesser extent to heavy main-line and terminal services. With larger production and closer attention to design features, cost and weights will tend to reduce. At the same time it is fairly certain that capacities will increase, which, together with multiple-unit operation, will render possible handling of the heaviest freight and passenger trains.

The total capital expenditure to be made in "Dieselizing" a railroad will be less in certain instances than with any form of electrification; this expenditure may be made gradually, and, by being spread out over a number of years, the financial strain sometimes necessitated when a complete electrification is undertaken will be avoided.

Synopses of Other Spring Meeting Papers

THE papers abstracted on this and the following page are being printed in full in pamphlet form for the Spring Meeting. Copies of them may be secured by filling out the blank on page 591 of this Mid-May issue of MECHANICAL ENGINEERING and mailing it to the office of the Society.

The remainder of the papers preprinted for the meeting will be found in this special Mid-May issue of MECHANICAL ENGINEERING.

The program for the Spring Meeting appears on page 501 of this issue.

High Steam Pressure and Temperature at Crawford Avenue Station

By ALEX D. BAILEY

Superintendent, Generating Stations, Commonwealth Edison Co., Chicago, Ill. Mem. A.S.M.E.

THIS paper discusses some of the problems encountered in the operation of the Crawford Avenue Station of the Commonwealth Edison Co. This station is designed for 550 lb. steam pressure at the turbines and 725 deg. fahr. temperature. The fuel used is central Illinois coal having an average heat value of 10,255 B.t.u. This is fired on forced-draft chain-grate stokers.

In the first three units the steam is reheated after passing through the high-pressure turbine by means of a special reheat boiler on each unit. The specified reheat temperature is 700 deg. The fourth unit has no reheat. The first three turbine units are bled at three points and the fourth at four. The boilers are of the interdeck superheater type, built by the Babcock & Wilcox Co., with a capacity of 150,000 to 200,000 lb. of steam per hour.

Unit No. 1, 50,000 kw., built by Parsons Co., England, was put into service in May, 1925. Unit No. 2, 60,000 kw., built by the General Electric Co., was put into service in November, 1924. Unit No. 3, 50,000 kw. built by the Westinghouse Co., was put into service in March, 1925. Unit No. 4, 75,000 kw., built by the General Electric Co., was put into service in August, 1926. Unit No. 5, 90,000 kw., built by the General Electric Co., is under construction, and will be put into service in August, 1927. No. 6, 100,000 kw., is on order with the Westinghouse Co., and will be in operation in the summer of 1928. Detailed descriptions of the station have been given in various reports and printed articles, so it is considered unnecessary for this discussion.

High Steam Pressures in Locomotive Cylinders

By LAWFORD H. FRY

Metallurgical Engineer, Standard Steel Works Company, Burnham, Pa. Mem. A.S.M.E.

IN THIS paper the author attempts an extended survey of the efficiencies obtainable with various steam pressures, and examines the effect of the ratio of expansion on the efficiency. The Rankine cycle, it is pointed out, does not offer a satisfactory basis of comparison for locomotives; therefore a modification is suggested, known throughout the paper as the "locomotive cycle," and all calculations of the paper are based on this cycle. Changes in boilers to permit operation at high pressures and temperatures are discussed, and it is pointed out that such a boiler would probably require some form of water-tube firebox. Detailed computations and comparisons of theoretical indicator diagrams are made, and the "locomotive cycle" is applied to various admission and release pressures. It is concluded that it is possible to secure a considerable increase in the thermal efficiency of the cylinders by increasing the boiler pressure. The use of three cylinders, one operating on high pressure and two on low, makes compounding a very simple matter, permitting the greatest return to be received from the high pressures. For the present, however, it is not felt to be expedient to use boiler pressures much in excess of 450 lb. per sq. in. gage.

The Study of Oil Sprays for Fuel-Injection Engines by Means of High-Speed Motion Pictures

By EDWARD G. BEARDSLEY

Junior Mechanical Engineer, National Advisory Committee for Aeronautics, Langley Field, Va.

APPARATUS for recording photographically the start, growth, and cut-off of oil sprays from injection valves has been developed at the Laboratory of the National Advisory Committee for Aeronautics at Langley Field, Va. The apparatus consists of a high-tension transformer by means of which a bank of condensers is charged to a high voltage. The controlled discharge of these condensers in sequence, at a rate of several thousand per second, produces electric sparks of sufficient intensity to illuminate the moving spray for photographing it. The sprays are injected from various types of valves into a chamber containing gases at pressures up to 600 lb. per sq. in.

Several series of pictures are shown. The results give the effects of injection pressure, chamber pressure, specific gravity of the fuel oil used, and injection-valve design, upon spray characteristics.

High-Pressure Steam at Edgar Station

By I. E. MOULTROP AND E. W. NORRIS

Respectively Chief Engineer, Construction Bureau, Edison Illuminating Company of Boston, and Engineer, Mechanical Division, Stone and Webster, Inc. Members A.S.M.E.

IN THIS paper the authors describe the design of a generating plant for the Edison Electric Illuminating Company using steam at 1200 lb. This plant, known as the Edgar Station, was the first in the country to use pressures of this magnitude. The equipment of the original plant is described and operating difficulties and results are discussed. An extension to the station is described and the results of experiences with the original plant as applied to the design of the extension are set forth. As in the case of the original installation, the plant layout and equipment are described and operating results given. The capacities of the boilers, turbines, and auxiliaries are given. An interesting discussion of the possibilities of pressures above 1000 lb. is included, as well as of some of the problems likely to be encountered in high-pressure-service equipment. The inadequacy of existing data is also mentioned. A comparison of the space requirements of high- and low-pressure equipment closes the paper.

Experimental Combustion Chambers Designed for High-Speed Diesel Engines

By CARLTON KEMPER

Junior Mechanical Engineer, National Advisory Committee for Aeronautics, Langley Field, Va.

IN THIS paper the preliminary requirement of the high-speed fuel-injection engine problem is given, and analyses of the cycles used in this type of engine are included. There is also given a discussion of the effect of increasing speeds on the output of engines employing these cycles. The requirements of combustion chambers are set forth and results of experiments using three different types are given in the paper. The engine and testing apparatus are fully

described and illustrated. Curves showing the performance of the engine when equipped with each of the special types of chambers also are shown.

Vibration of Frames of Electrical Machines

By J. P. DENHARTOG

Research Engineer, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

IN CONNECTION with the attempts to reduce the noise of electrical machinery in general, and more especially of small motors for domestic use, it is of importance to be able to calculate the natural frequency of the component. It is shown that the frame, which usually emits a large portion of the total noise, can, in may cases, be regarded as a part of a ring with rigid ends. Formulas and curves are given in the paper for the calculation of this fundamental frequency. The derivation of these formulas is given in an Appendix.

Economics of Coal Carbonization in the United States

By GEO. A. ORROK

Consulting Engineer, New York, N. Y.

Paper covers a review of the progress of carbonization in America, bringing it down to date.

Contributors to This Issue

WILLIAM ARTHUR

William Arthur, manager of the traction division, American Brown Boveri Electric Corporation, Camden, N. J., was born in Wisbeach, England. He was educated at Horwich, Lancashire Railway Technical Institute, and Liverpool University. He served his engineering apprenticeship in works in Horwich from 1893 to 1899, and then became assistant engineer in the electric traction department of the Lancashire and Yorkshire Railway. Three years later he was appointed assistant railway electrical engineer in charge of the Liverpool-Southport electrified system.

In 1908 Mr. Arthur came to the United States and entered the railway department of the Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. In 1911 he became assistant engineer of the New York, New Haven & Hartford electrified system under W. S. Murray, and the following year entered the consulting field with the firm of McHenry & Murray, engineers of New Haven, Conn., where he remained for four years. He then opened his own office as consulting engineer for electric railways. Mr. Arthur is president of the Arthur Power-Saving Recorder Co., and vice-president and director of the Persons-Arter Machine Co., Worcester, Mass.

Mr. Arthur has been granted numerous patents for controlling arrangements for electric trains, for power-saving devices for electric railways, and for various other inventions. He did splendid work during the World War in conservation of coal and power in the United States and Canada. He originated the means for eliminating low-power factors and decreasing line losses in alternating-current railway systems.

Mr. Arthur is the author of many articles and reviews published in the technical press. He is a member of the American Institute of Electrical Engineers, of the American Electric Railway Association, and of several clubs.

* * * * *

L. C. BELL

Landon C. Bell is assistant general counsel secretary, and director of the W. M. Ritter Lumber Co., Columbus, Ohio, with which concern he has been identified for twenty years. He was educated at Milligan College and the University of Virginia, and holds the degrees of Ph.B., M.A., and LL.B. He is a life member of the Virginia Historical Society and of the Ohio Historical and Archaeological Society, and belongs also to a number of clubs.

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W. S. CONANT

W. S. Conant is a consulting engineer of Washington, D. C., as well as a lieutenant-colonel in the Ordnance Reserve Corps. He specializes on plans and specifications of steam and electric plants; and has taken out a number of mechanical patents, including one for a system of air-humidity control. He was educated at St. Paul's School at Concord, N. H., and the Lawrenceville School in New Jersey, and received his degree of A.B. in 1890 and E.E. in 1892 from Princeton University. In 1917 he was commissioned captain of the Ordnance Reserve Corps, and served as assistant to the officer in charge of shops and later as construction officer at the Springfield Armory in the manufacture of service rifles. In 1918 he was made major, and in 1924 was promoted to the rank which he now holds.

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WM. S. LEE

William States Lee is vice-president and chief engineer of the following companies: Southern Power Co., Great Falls Power Co., Western Carolina Power Co., Wateree Power Co., Duke Power Co., Catawba Manufacturing and Electric Power Co., Catawba Power Co., Duke-Price Power Co., Ltd., and the Quebec Development Co., Ltd. He is president and chief engineer of the Piedmont and Northern Railway Co., and consulting engineer for the Chute-a-Caron Power Co., Ltd., Aluminum Co. of Canada, Ltd., and the Tallassee Power Co. He is Trustee of the Duke Endowment, director of the American Cyanamid Co., inventor of the Lee pin, and a pioneer in high-tension hydroelectric power development and transmission. His offices are located in Charlotte, N. C., New York, N. Y. and Montreal, Canada. Mr. Lee holds membership in the A.S.M.E., A.S.C.E., Engineering Institute of Canada, and the North Carolina Society of New York, and is also a Fellow of the A.I.E.E.

H. M. JACKLIN

H. M. Jacklin, assistant professor of automotive engineering at Ohio State University, is a graduate of the Michigan State College, East Lansing, Mich., from which school he received his bachelor's degree in 1913 and his professional degree in 1919. He has been connected with the Novo Engine Co., the University of Wisconsin, and the Oneida Motor Truck Co., and has been at Ohio State University since 1921.

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L. C. JOSEPHS, JR.

L. C. Josephs, Jr., is in charge of engineering at the Mack Plant of the International Motor Co., Allentown, Pa., where trucks, buses, bodies, and various components are made. Since graduation from Harvard in 1908, Mr. Josephs has been continuously connected with transportation work, principally in the manufacturing of equipment. After an apprenticeship at the Westinghouse Co. and a short period with the Pennsylvania Railroad, he served in the Electric Locomotive Department of the General Electric Co. at Schenectady until 1917, when he left to join the Army. During the War he served in several capacities in the Army Engineers, finally in charge of engineering work on searchlights in the office of the Chief of Engineers. After the War he joined the engineering organization of Mack Trucks.

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Bachrach

J. F. LINCOLN

J. F. Lincoln is president of the Lincoln Electric Co., and director of the Reliance Trust Co. He was graduated from Ohio State University in 1907 with the degree of E.E. From college he entered the employ of the Lincoln Electric Co., and seven years later was promoted to the position of general manager. He has been very active in the development of both arc welding and the application of arc welding to manu-

facturing work, and is the inventor of the adjustable-speed motor of the shifting-armature type manufactured by the Reliance Electric and Engineering Co., on which the amount of speed-range adjustment is unlimited. The system of bonding and track welding used by the Rail Welding & Bonding Co. is also his invention.

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RAY S. QUICK

Ray S. Quick was graduated from the University of California in 1916 with the degree of B.S in E.E. In 1919 he received his M.S. in E.E. from the University of Illinois following a research fellowship in the department of electrical engineering at the Engineering Experiment Station of that institution. From October 1919 to the present date he has been connected with the Pelton Water Wheel Co., San Francisco, Cal. During the War Mr. Quick served as lieutenant in the Signal Corps of the U. S. Army.

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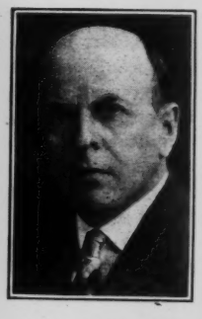

ARNOLD PFAU

Arnold Pfau, consulting engineer of the hydraulic department of the Allis-Chalmers Manufacturing Co., Milwaukee, was graduated from the Swiss Polytechnic of Zurich in 1896. He designed hydraulic turbines for Escher-Wyss & Co., of Zurich, until 1900, when he became chief engineer of the hydraulic department of Ruston & Co., Vienna. In the spring of 1904 he was reëmployed by Escher-Wyss & Co. to introduce turbines at the Allis-Chalmers Works in Canada and the U. S. A. Mr. Pfau is also the founder of the American Resistor Co., now the American Resistor Corporation, Milwaukee, Wis. From 1905 to 1908 he was chief engineer of the hydraulic department of the Dixon Works of the Allis-Chalmers Co., at Scranton, Pa., and the Reliance Works at Milwaukee. Since 1908 Mr. Pfau has been a consulting engineer in the field of turbine design.

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Frederick E. Searle, arthur of the paper entitled A Description of the Henry Ford Trade School, holds the position of superintendent of Ford Schools.

ALBERT SOBEY

Albert Sobey received his B.S. in mechanical and electrical engineering at the Michigan Agricultural College in 1909, and during that year was employed by the Michigan Power Co. and the W. H. Zimmerman Co., Lansing, Mich. In 1910 he was placed in charge of the Chicago office of the Zimmerman Co. He held an instructorship in physics, electrical engineering, and mathematics at the Michigan College of Mines from 1912 to 1918, and the following year served as assistant chief and chief of the Radio Service Division of the Military Intelligence Division of the U. S. A., holding the rank of captain and later that of major. Upon leaving the Army in 1919 he became director of education of the Industrial Mutual Association, in charge of educational work for the allied factories of Flint.

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J. HALL TAYLOR

J. Hall Taylor, president of the American Spiral Pipe Works, Chicago, Ill., studied engineering at Lewis Institute, Chicago. In 1900 he organized the above company and later developed the method of forging seamless-steel boiler nozzles and pipe flanges. He has devoted the last 27 years to handling all of the engineering and development work of this firm.

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E. O. WATERS

E. O. Waters received his technical education at the Sheffield Scientific School of Yale University from which he was graduated with the degree of M.E. in 1914. From 1914 to 1918 he held the position of instructor in machine design at his alma mater. During 1918 he served in the office of the Chief Ordnance Officer, A. E. F., in connection with the construction and maintenance of repair shops in France for ordnance material. Since returning to Yale University in 1919, he has held the position of assistant professor of mechanical engineering in charge of machine design. Professor Waters has been a frequent contributor to the technical press. During the summer months he has spent considerable time in the engineering and test departments of the Winchester Repeating Arms Co., the Cincinnati Milling Machine Co., the General Electric Co., and various other concerns.

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CHARLES P. WOOD

Charles P. Wood, engineer in charge of special investigations and reports with Lockwood, Greene & Co., Inc., New York, was graduated from Cornell University in 1904 with the degrees of M.E. and E.E. Until 1917 Mr. Wood was engaged in general operating, designing, and contracting work in the South and the East on cotton-oil mills, steam and gas-power plants, brick and tile plants and by-product recovery plants.

During the War he served with the 30th Engineers, A. E. F., later known as the First Gas Regiment. He was promoted from the rank of lieutenant to that of major and was awarded the Croix de Guerre by the 32nd Corps of the French Army. After the War he became special trade commissioner with the U. S. Department of Commerce, making a report on industrial machinery in France and Belgium. Since 1920 he has been associated with Lockwood, Greene & Co.

Mr. Wood is a member of the American Institute of Electrical Engineers, the Society of American Military Engineers, Cornell Society of Engineers, the New York State Society of Professional Engineers, Military Order of Foreign Wars, the American Legion, and the Sons of the American Revolution. He belongs also to a number of clubs.

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J. K. WOOD

Joseph K. Wood received his engineering education in Cooper Institute of Technology. As a consulting engineer on spring design and engineering materials, Mr. Wood has been engaged by both the Westinghouse Electric & Manufacturing Co. and the Western Electric Co. Mr. Wood served as an officer in engineering service during the War. In 1923 he was appointed chairman of the A.S.M.E. Special Research Committee on Spring Design.

The following papers, to be presented at the White Sulphur Springs Meeting, will be available in pamphlet form and may be secured without cost by any member of the Society.

Check papers desired and mail to the Secretary.

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	FRY, L. H., High Steam Pressures in Locomotive Cylinders (available May 20)		
	KEMPER, C., Experimental Combustion Chambers Designed for High-Speed Diesel Engines (now available)		
	MOULTROP, I. E., and NORRIS, E. W., High-Pressure Steam at Edgar Station, Boston (available May 20)		
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